AD

A SURVEY OF TECHNOLOGY FOR HYBRID VEHICLE AUXILIARY POWER UNITS

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By

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13. ABSTRACT (Maximum 200 words)

The state-of-the-art of heat engines for use as auxiliary power units in hybrid vehicles is surveyed. The study considers reciprocating or rotary heat engines, excluding gas turbines and fuel cells. The relative merits of various engine-generator concepts are compared. The concepts are ranked according to criteria tailored for a series-type hybrid drive.

The two top APU concepts were the free-piston engine/linear generator (FPELG) and the Wankel rotary engine. The FPELG is highly ranked primarily because of thermal efficiency, cost, producibility, reliability, and transient response advantages; it is a high risk concept because of unproven technology. The Wankel engine is proven, with high power density, low cost and low noise. Four additional competitive concepts include two-stroke spark-ignition engine, two-stroke gas generator with turboalternator, free-piston engine gas generator with turboalternator, and homogeneous charge compression ignition engine. This study recommends additional work, including cycle simulation development and preliminary design to better quantify thermal efficiency and power density.

Auxiliary concepts were also considered, including two which warrant further study: electrically actuated valves, and lean turndown of a normally stoichiometric engine. These concepts should be evaluated by retrofitting to existing engines.

14. SUBJECT TERMS

Heat Engine	Diesel	Turboalternator
Rotary Engine	Spark-Ignition	Linear Generator
Hybrid Vehicle	Two-Stroke	Gas Generator
Auxiliary Power Unit	Four-Stroke	HCCI
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EXECUTIVE SUMMARY

Problems: This study was undertaken to survey the state-of-the-art of heat engines for use as power plants in hybrid vehicles. It assumes that the heat engine drives an electric generator providing auxiliary power for charging batteries and/or powering the electric traction motor, which is the primary drive of the vehicle. The study is confined to reciprocating or rotary heat engines, excluding gas turbines and fuel cells.

Objective: The objective of this project was to survey the state-of-the-art of heat engines for use as powerplants in hybrid vehicles.

Importance of Project: While this study is useful, it is necessarily subjective due to the lack of consistently defined quantitative information on engines in the power class needed for a hybrid APU and on advanced concepts. The ranking study is intended to narrow the focus of research by eliminating concepts that are not likely to succeed in the hybrid APU application, and by focusing attention on those parameters that need to be further quantified. The recommended next step is in-depth analysis of those concepts that offer the most promise based on this study.

Technical Approach: A literature survey was performed to determine the relative merits of various engine-generator concepts. The concepts were ranked according to criteria tailored for a series-type hybrid drive. The ranking procedure assigned weights to each criterion according to its relative importance in hybrid APU applications. By this method, it is hoped that those concepts unlikely to be competitive are systematically eliminated, and those concepts most deserving of further study and development are highlighted.

Accomplishments: The two most promising APU concepts were the free-piston engine/linear generator (FPELG) and the Wankel rotary engine. The FPELG is highly ranked primarily because of perceived advantages in thermal efficiency, cost, producibility, reliability, and transient response; however, it is a high risk concept because of unproven technology for the generator. The Wankel engine is a proven concept with high power density and benefits of relatively low cost and noise. Four additional concepts ranked somewhat lower but within the range of competitiveness for this subjective analysis. These include two-stroke spark-ignition engine, two-stroke gas generator with turboalternator, free-piston engine gas generator with turboalternator, and homogeneous charge compression ignition engine. This study recommends additional evaluation of these concepts, including cycle simulation work and preliminary design to better quantify thermal efficiency and power density.

Auxiliary concepts were also considered, which include those ideas that are not specific to a given engine design but may be applied to a number of different engine types. Of these, two stood out as warranting further study: electrically actuated valves, and lean turndown of a normally stoichiometric engine. It is recommended that these concepts be evaluated by retrofitting to existing engines.

Military Impact: Identification of alternative power plants for hybrid electric vehicle application provides options for military selection. The most optimum engine for military use would be determined from cost benefit and trade-off studies, commercial availability, and requirements for technology demonstration. Of major significance is whether the hybrid electric vehicle drive would be used for administrative-tactical wheeled vehicle or combat-tracked vehicle applications.

FOREWORD/ACKNOWLEDGEMENTS

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I. INTRODUCTION

The Advanced Research Projects Agency (ARPA) has taken an initiative in developing hybrid vehicle technology, focusing on the unconventional technologies needed to make hybrid vehicles successful as energy-saving and pollution-reducing alternatives to current automotive technology. The model hybrid electric drivetrain consists of an auxiliary power unit (APU) comprising a combustion engine driving an electric generator, an electric power storage device (battery), electric traction motors, and a control system. The APU must provide electric power under conditions very different from the conventional automotive driving cycle and must meet a different set of objectives. It is natural, then, that the power plant itself may be unconventional. This study explores the alternative combustion engine power plants that may be used for hybrid electric vehicles. The intent is to discriminate between both conventional and unconventional alternatives to find those concepts that are most worthy of further study in the form of simulation, analysis, and ultimately, prototype demonstration.

The study was initiated as a brainstorming exercise among a number of engineers at Southwest Research Institute (SwRI) involved in engine design and development, and in hybrid vehicle development. The author then proceeded to review the literature available on conventional and unconventional engines and develop a list of concepts, which were then assessed and ranked according to their perceived ability to meet the objectives of a hybrid APU. The concepts, criteria, and ranking are discussed in this report, and recommendations for further study are provided.

II. OBJECTIVE

The objective of this study is to discern which concepts for combustion engines are the best choices for detailed study as alternative power plants to the conventional reciprocating engine in application to hybrid electric vehicle APUs. The study is limited to internal combustion engines or reciprocating external combustion (Stirling) engines suitable for coupling to electric generators; turbine engines, fuel cells, and other concepts are also viable alternatives but are not considered

here. The criteria are tailored for series-type hybrid drivetrains in which the combustion engine drives electric generation as a supplemental charging source for battery storage. Other types of hybrid drives are possible, such as a parallel arrangement whereby the combustion engine directly drives the traction wheels through a transmission in combination with electric motors (the criteria for these systems will be markedly different).

III. CRITERIA

Hybrid powertrains are considered alternatives to conventional combustion engines for two primary reasons: fuel economy and emissions. The fuel economy benefit arises from two fundamental factors: power leveling by energy storage in the batteries, and the ability to operate the APU power plant at constant, peak efficiency conditions. Emissions are improved because the overall fuel consumption is low, allowing the engine to operate at a near steady state cycle. Thus, the criteria for selection of an engine for a hybrid vehicle are very different from those for a conventional automotive power plant. The criteria that are important to hybrid APUs are listed below, roughly in their order of precedence.

A. Power Density

The volume available for the power plant is limited, as space must be allotted for energy storage, traction motors, generator, and controls, as well as the engine itself. The weight is important, since vehicle weight directly impacts fuel economy. High power density, in terms of both power-to-volume ratio and power-to-weight ratio, is desirable. Power density in consideration of both engine and generator tends to favor high-speed engines, as the generator volume and weight for a given power rating decreases with increasing speed.

B. Emissions

If an engine has good power density but substantially worse emissions characteristics than its conventional competitor, it ultimately will not fulfill the objectives of hybrid vehicles. Thus,

concepts that improve power density at the expense of emissions will not be completely satisfactory. However, brake-specific emissions that are higher than the conventional technology may be tolerable as the duty factor for the APU is much lower than the conventional drivetrain.

C. Thermal Efficiency

Thermal efficiency--a measure of fuel economy--must be high to achieve overall vehicle fuel economy benefits over conventional drivetrains. While the hybrid drivetrain has key advantages, it must also overcome losses associated with energy storage and retrieval, and generator and motor inefficiencies, as well as the simple problem that the package generally tends to be heavier than a conventional drivetrain.

D. Cost

To be competitive as a commercial product, a hybrid vehicle must have comparable cost to conventional vehicles. This equation will have to factor in total life cycle costs, which include reduced fuel cost as well as the expense of battery replacement and system maintenance. It is clear that the drivetrain will be significantly more expensive due primarily to the energy storage system. Thus, the engine itself will have to be as inexpensive as possible.

E. Transient Response

Transient response characteristics of a hybrid APU are a lower priority, as the system is designed to operate at near steady state conditions. However, a rapid transient response is desirable for rapid engine catalyst warm-up, assuming a catalyst is used. The response of most interest is the start-up from cold to full power.

F. Producibility

Closely linked to the cost issue is the producibility of the engine. Will it be manufactured by conventional methods, or will new or expensive, unconventional processes be required?

G. Reliability

Also ultimately a life cycle cost issue, the reliability of combustion engines is a factor to consider. Many small power plants for utility applications are designed for much shorter life than their automotive counterparts. However, engine replacement or major overhaul should not be any more frequent for hybrid powertrains than for conventional engines. The design of APU engines for reliability must take into account the high power factor. The APU will be operated primarily at its design point, instead of primarily at low load conditions like a conventional automotive engine.

H. Cranking Torque

During start-up, most engines absorb a considerable amount of power from the starter motor to overcome friction and inertia. This start-up energy can be characterized as the cranking torque and is highly dependent upon engine design. It is a direct penalty to overall thermal efficiency and must be taken into consideration for drivetrains where the APU is cycled frequently between running and nonrunning conditions.

I. Noise, Vibration, and Harshness

Noise, vibration, and harshness (NVH) are measures of the comfort level of passengers in vehicles or of bystanders. These factors are increasingly important for customer satisfaction and acceptance. In this regard, the hybrid drivetrain should not be appreciably worse, and preferably should be better than its conventional counterpart.

J. Technical Risk

The technical risk of a new concept is not a measure of how well it will perform its intended function, but of how much is unknown about its viability, and how much research effort may need to be expended to make it viable.

K. Multifuel Capability

Some engine concepts are limited to certain types of fuel. Alternative fuels such as natural gas and methanol are of increasing interest primarily due to environmental concerns. The ability to be tolerant of different fuels would be an advantage.

IV. RANKING PROCEDURE

As a systematic means of distinguishing between APU concepts, a ranking procedure was applied. In this procedure, each concept was assigned a score relative to each of the previously described criteria. The criteria were also weighted according to their importance in the APU application. Where possible, the assigned scores were based on actual quantifiable data. Thermal efficiency is defined as the ratio of electric power out to fuel energy in. The power density for an engine type is scored as an actual (reciprocal) specific weight in kW/kg, or specific volume in kW/L (the reciprocal is used to conform to the overall strategy of "higher is better"). For parameters which have no quantifiable level, a relative score was assigned with a baseline level of 1.0. Weighting factors were assigned as the product of a normalizing factor established as the reciprocal of the difference between maximum and minimum values of the parameter, and a significance weight in the range of 0 through 5.

V. STATE-OF-THE-ART

As part of this study, an extensive literature review was conducted to assess the current state-of-the-art for automotive combustion engines and particularly for hybrid APUs in relation to the above criteria, and to examine the potential for advanced concepts to improve the state-of-the-art. It is important to understand the best attainable performance of a conventional engine-generator combination in order to assess the value of unconventional technologies in improving this performance. During this study, a total of 58 references were reviewed. Supplementary

information was obtained from the SwRI engine database, which is a comprehensive listing of published data on reciprocating engines from all over the world.

One key reference is a similar study that was conducted eleven years ago by JPL for the U.S. Department of Energy (1)*, written by Mr. H.W. Schneider. The objectives of that study were somewhat different in that the primary focus was for parallel drivetrains; however, the document fairly summarizes the state-of-the-art for that time, and also makes recommendations for series drivetrains. The author considers in his discussion the following alternatives:

- Four-stroke piston engines
- Four-stroke rotary engines (Wankel)
- Two-stroke piston engines
- Indirect injection diesel engines
- Direct injection diesel engines
- Brayton cycle engines (gas turbine)
- Stirling engines
- Rankine cycle engines (steam engine)
- Free-piston engines.

After consideration of the relative merits for series drivetrains, the author recommended research to be prioritized as follows: direct-injected spark-ignited two-stroke engines, fuel-injected rotary engines, and free-piston engines.

This study was revised and updated in 1992 by A.F. Burke, with special consideration of the driving cycle requirements for a series hybrid drivetrain.(2) The developments in the years between 1984 and 1992 had made the series option more feasible, primarily because of advancements in electric drivetrain component sizes and efficiencies. For a series drivetrain application to a passenger car, gradability and minimum sustained highway speed requirements dictate an APU of about 25 to 30 kW output, suggesting that an appropriate target power rating

* Underscored numbers in parentheses refer to the list of references at the end of this report.

for the engine is 30 kW. Mr. Burke places a lower priority on fuel economy for the combustion engines in a series drivetrain because the vehicle is assumed to be operating in a purely electric mode for 80 to 90 percent of the driving miles, based on results of a Monte Carlo analysis of typical driving patterns.(3) For the series hybrid, start-stop operation is also less of a consideration than for the parallel configuration because the periods of continuous operation and continuous non-operation are longer. According to Mr. Burke's analysis, the series drivetrain may go for days at a time without using the heat engine due to short urban trips with battery charging from the wall plug between trips. In a similar study, Mr. Burke focussed on the effects of start-stop operation indicating that for range extender type operation, where the vehicle would be operated for extended periods in highway driving with fairly low, continuous power demand, the overall fuel economy could be greatly improved if the APU were operated intermittently at its peak efficiency rather than continuously at low load.(4) These updates redefined the APU engine development priorities as 1) rotary, 2) direct injection two-stroke, and 3) gas turbine engines, omitting discussion of Stirling, Rankine, or free-piston engines.

The survey of literature done by Mr. Schneider and updated by Mr. Burke distills the state-of-the-art into estimates of performance and power density attainable by the various current technology engine types. These summaries are included in the Appendix, which also incorporates data from other sources as described below. Figure 1 shows the projections of specific weight for the conventional technology engines, as presented by these two references.

In the legend, "2-S SI" refers to two-stroke spark-ignition engines, "Rotary" to Wankel-type rotary engines, "4-S SI" to four-stroke spark-ignition engines, "4-S DI CI" to four-stroke direct injection compression ignition (Diesel) engines, and "4-S IDI CI" to four-stroke indirect injection compression ignition (Diesel) engines. The later reference applies a bit more pessimism to the two-stroke gasoline engines but still ranks them at the lowest specific weight. Schneider presented both DI and IDI diesel engines but discounted them as contenders for the APU due to high specific volume and weight; Burke did not consider these engines. Burke was more optimistic for rotary and four-stroke spark-ignition engines. Figure 2 shows the comparison for specific volumes, which follows similar trends to specific weight. However, Burke was more

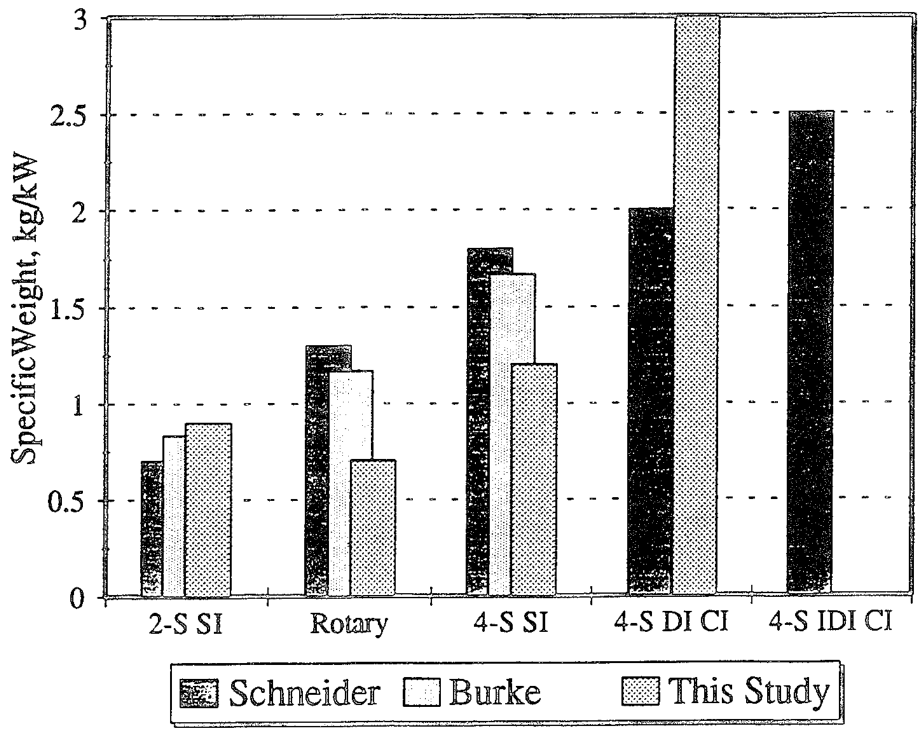


Figure 1. Specific weight comparisons from References 1 and 2

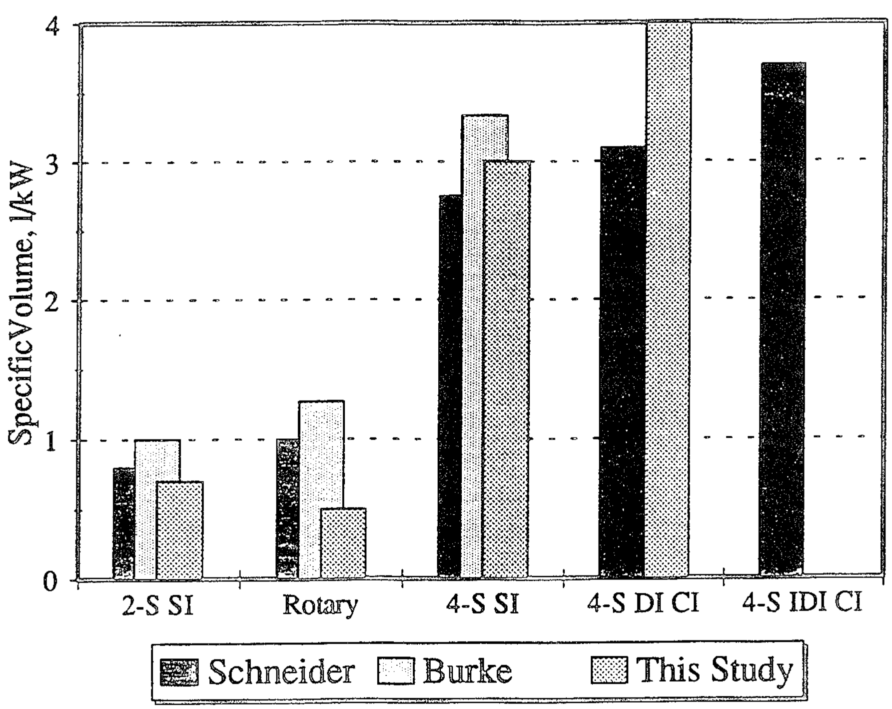


Figure 2. Specific volume comparisons from References 1 and 2

pessimistic than Schneider in his predictions. In terms of fuel consumption (see Fig. 3), the two-stroke spark-ignition engine was a strong contender, with fuel economy approaching or exceeding that of the IDI diesel. Fuel economy (along with simplicity) was a key reason for Schneider's recommending the two-stroke SI engine. Burke's update has added some pessimism to this prediction, but this engine still appears to be a strong contender in his analysis. Of course, the DI diesel scores best in fuel consumption, but it is still discounted by these researchers because of its perceived weight and volume penalties.

Reference 5 analyzes key driving conditions of acceleration, gradability, and steady state fuel economy and compares alternative engines in application to a range-extender vehicle (REV) based on an existing Ford Taurus platform (two other platforms were analyzed but not explicitly discussed in the reference). This study determined that the power rating for the APU engine should be in the range of 25 to 50 kW for this type of application. Several alternative engines were considered as substitutes for a small (6.5-kW) genset that was considered in a previous

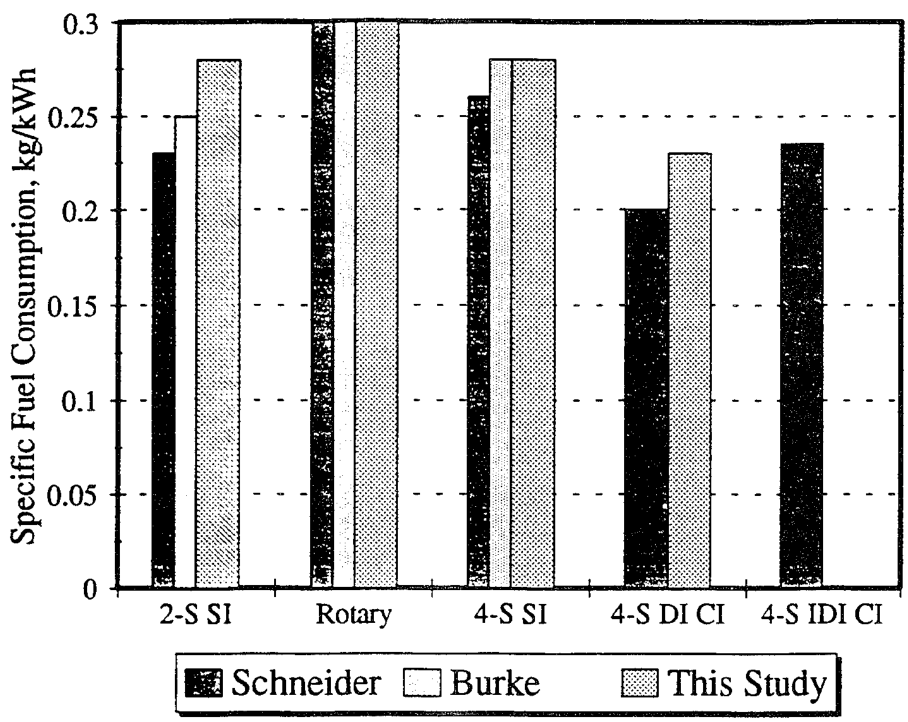


Figure 3. Fuel consumption comparisons from References 1 and 2

phase of that analysis. The alternatives included the Orbital OCP "X" engine rated at 37.6 kW, the GM Quad 4 four-stroke SI engine rated at 62.7 kW, the Nomac gas turbine genset at 24.0 kW, the NASA Series 70 rotary engine at 28.1 kW, and an MTI Mod II Stirling engine at 32.2 kW. The comparison criteria based on vehicle performance with these options are not very useful in distinguishing between alternative engine categories because the power ratings of the different engines vary widely. Obviously, acceleration and gradability criteria favored the larger power plants, whereas fuel economy favored the Stirling engine. This reference does provide a useful comparison of performance data for several state-of-the-art engine categories, which is included in the Appendix.

The current state-of-the-art for conventional technologies is discussed in the sections that follow. In this context, conventional technologies are those currently in production for automotive engines, or in imminent production status. Such technologies as free-piston engines or Stirling engines are still very much in the research stage and will therefore be considered as advanced concepts.

A. Generator Technology

Since the objective is electric power out, the generator characteristics must be considered as well as the engine characteristics. The power density of conventional rotating generators is significantly enhanced by increasing the shaft speed. The efficiency is less affected, as proper design can achieve high efficiency at virtually any design speed. At higher speeds, windage losses tend to degrade the efficiency somewhat; however, for this study, it is assumed that efficiency is constant with speed. For those concepts that work with a conventional rotating generator, it is assumed that the generator efficiency is 92 percent. The overall design requirements of a hybrid APU favor those engines that have higher shaft speed for compactness of both engine and generator. Figure 4 shows the characteristic tradeoff of generator size and weight with shaft speed for a 25-kW generator.(6) For a baseline unit operating at 6,000 rpm and generating 30 kW, specific volume and weight are assumed to be 0.3 L/kW and 1.5 kg/kW, respectively. These values are additive to engine specific volume and weight to obtain system

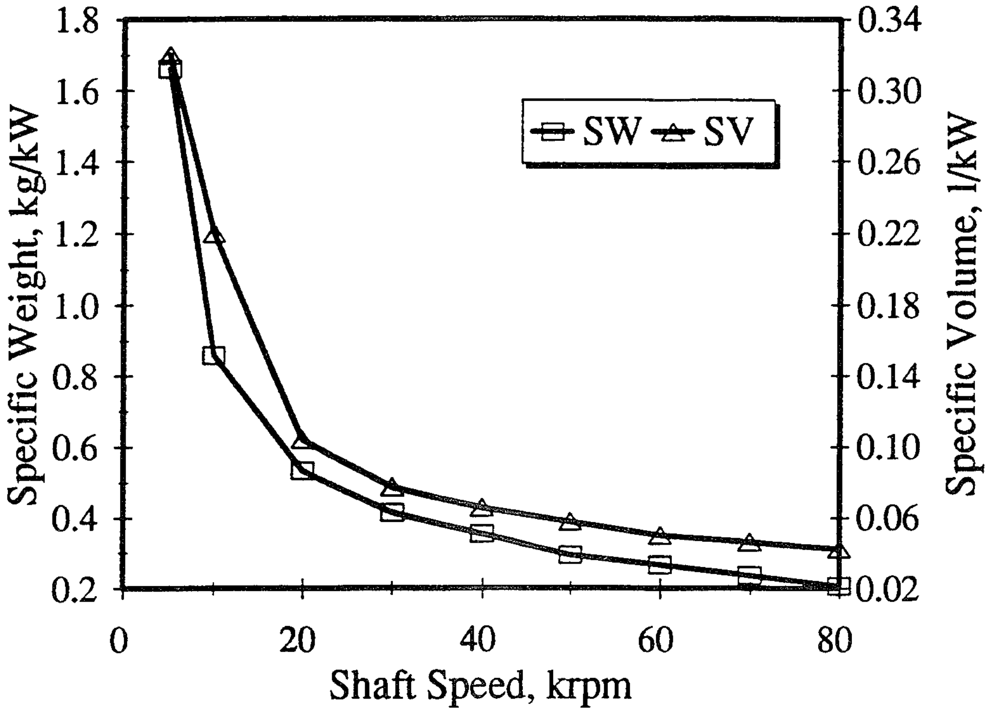


Figure 4. Generator size and weight as a function of shaft speed

characteristics. For systems which afford higher shaft speeds, these values are reduced according to the relationships shown in Fig. 4.

B. Four-Stroke Spark-Ignited Engines

The most highly developed and sophisticated reciprocating engines currently in wide use are the automotive four-stroke spark-ignited engines. Most domestic automobiles employ some version of this class of engine, burning gasoline. The characteristics of the current four-stroke SI are as follows:

- Overhead cams, frequently separate intake and exhaust cams
- Four valves per cylinder
- Electronic fuel injection at the intake port
- Electronic spark control
- Sophisticated digital feedback control systems
- Tuned induction systems
- Variable valve timing (in advanced engines)

- Variable intake geometry (in advanced engines)
- Lightweight materials (aluminum, plastics, composites).

Most of these technologies have not yet been incorporated in engines of the size class appropriate to hybrid APUs. Turbocharging was a common feature of automotive engines in the mid-1980s but has gradually disappeared due primarily to durability issues and cost, the advantages in power density being largely displaced by the improvements due to other technologies. The durability problems with turbocharging the SI engine are attributed to high turbine inlet temperatures, since these engines run with stoichiometric fuel-air ratios. Figure 5 (reproduced from Reference 7) illustrates the trends in power density based on ninety 1988 passenger car engines. Those engines incorporating multivalve technology approach the turbocharged engines in power density. It seems unlikely that turbocharging will be a feasible option to obtain high power density on an SI engine of the power class needed for a hybrid vehicle APU. The remaining technologies previously listed are all feasible for consideration in the development of the hybrid APU.

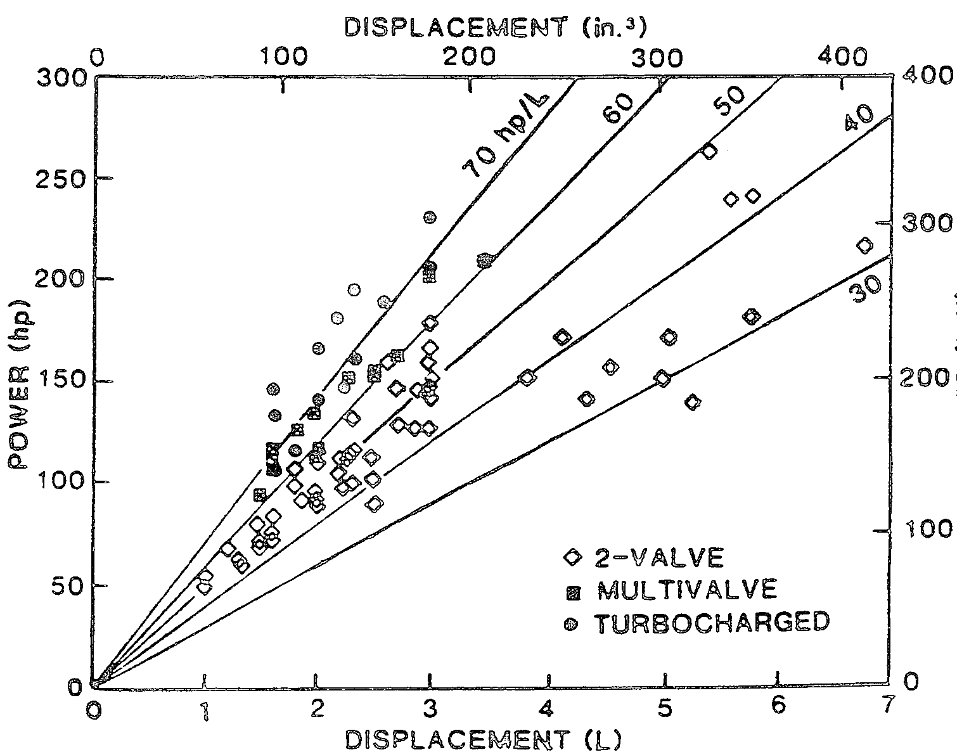


Figure 5. Power density trends from Reference 7

TABLE A-1 in the Appendix summarizes the state-of-the-art of automotive SI gasoline engines, based on the current literature. Production engines are shown along with racing engines to illustrate the range of performance attainable by applying higher technologies.

Figure 6 shows the state-of-the-art for specific weight for four-stroke spark-ignition engines. There is a noticeable increase in specific weight as the power drops below 50 kW into the size class appropriate to APUs. The best obtained specific weight for a 30-kW engine is about 1.9 kg/kW, slightly higher than values suggested by Schneider and Burke. However, the significant sensitivity to power rating in this range (and the lack of engines at 30- to 40-kW rating) suggests that this level may be challenged by focused effort on this application using existing technologies. The higher specific weight of engines below 30-kW output is likely attributable to the lack of penetration of the latest automotive technologies listed above. It is not unreasonable to presume that a small (30-kW) engine incorporating the latest technologies could achieve specific weight of 1.2 kg/kW.

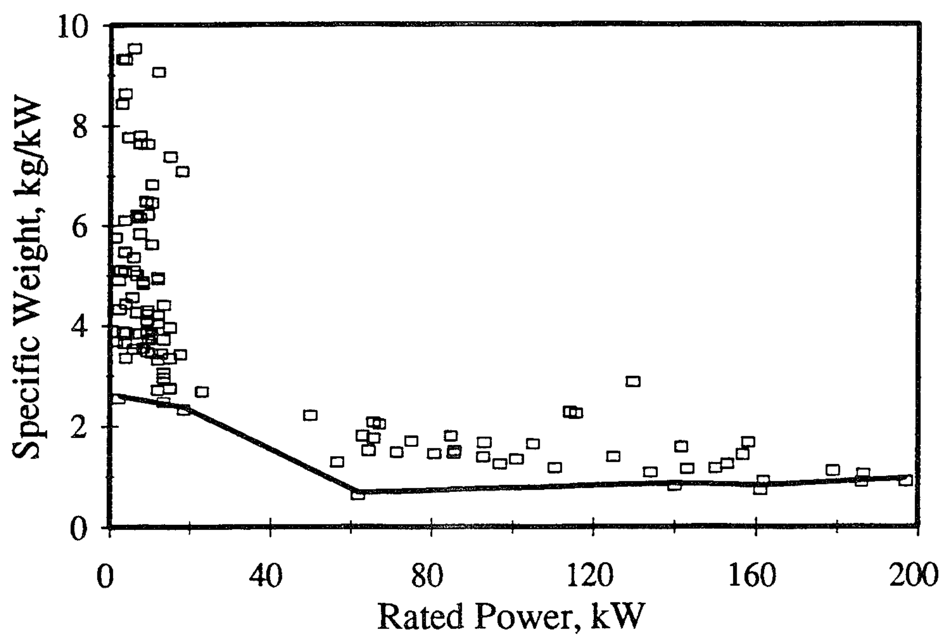


Figure 6. <u>**Specific weight of four-stroke spark-ignited engines**</u>

Specific volume for four-stroke spark-ignition engines is shown in Fig. 7. Again, there is a substantial increase in the state-of-the-art specific volume as the rated power drops below 50 kW. The best obtained specific volume for an engine 30 kW or less is about 3.6 L/kW, higher than the data of Schneider and Burke. Challenging this level will be more difficult than the specific weight objective, as some of the advanced technologies for weight reduction and power increase have no impact or adverse impact on package volume. A specific volume of 3.0 L/kW is probably achievable by a four-stroke SI engine developed specifically for the APU application.

Fuel consumption of four-stroke SI engines is rarely reported in the literature in the form used for comparison by Schneider and Burke (i.e., BSFC), as these engines are generally more concerned with fuel economy (i.e., miles per gallon) on vehicle driving cycles. Nevertheless, the limited data available from the literature are shown in Fig. 8. Best BSFC from this chart is about 0.26 kg/kW·h, consistent with the data of Burke. It may be surmised that for lower output engines, BSFC will be somewhat higher still, as friction and heat transfer losses proportionally increase with decreasing engine displacement. Best BSFC for a 30-kW engine is probably about 0.28 kg/kW·h.

Emissions from four-stroke SI engines are rarely reported in the literature. Even if they were, the comparisons would probably not be useful for the APU application because emissions are generally measured on a vehicle driving cycle that would be entirely different from the operating cycle of the APU. The state-of-the-art for emissions control is well developed and is well represented by the current legislated emissions limits, as engine manufacturers generally calibrate their engines to just meet these limits with adequate margin for variability. Four-stroke spark-ignition engines for vehicle applications almost universally use a three-way catalyst for control of emissions of oxides of nitrogen (NO_x), hydrocarbons (HC), and carbon monoxide (CO) in combination with a feedback control system to manage the engine air-fuel ratio within very tight limits around the stoichiometric condition. The feedback control uses an oxygen sensor in the exhaust to detect the presence of oxygen (O_2), which indicates a lean condition. A few engines are applying lean-burn technology with stratified charge to improve fuel consumption; in this case, wide-range O_2 sensors are used to apply feedback control over a wider stoichiometry range,

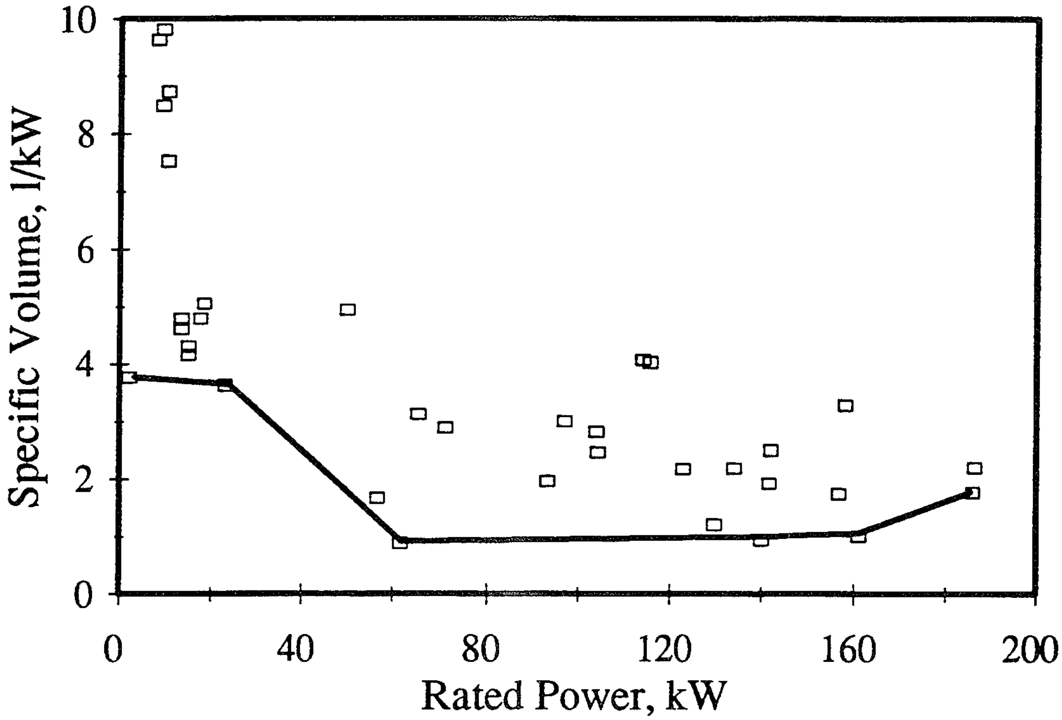


Figure 7. Specific volume of four-stroke spark-ignited engines

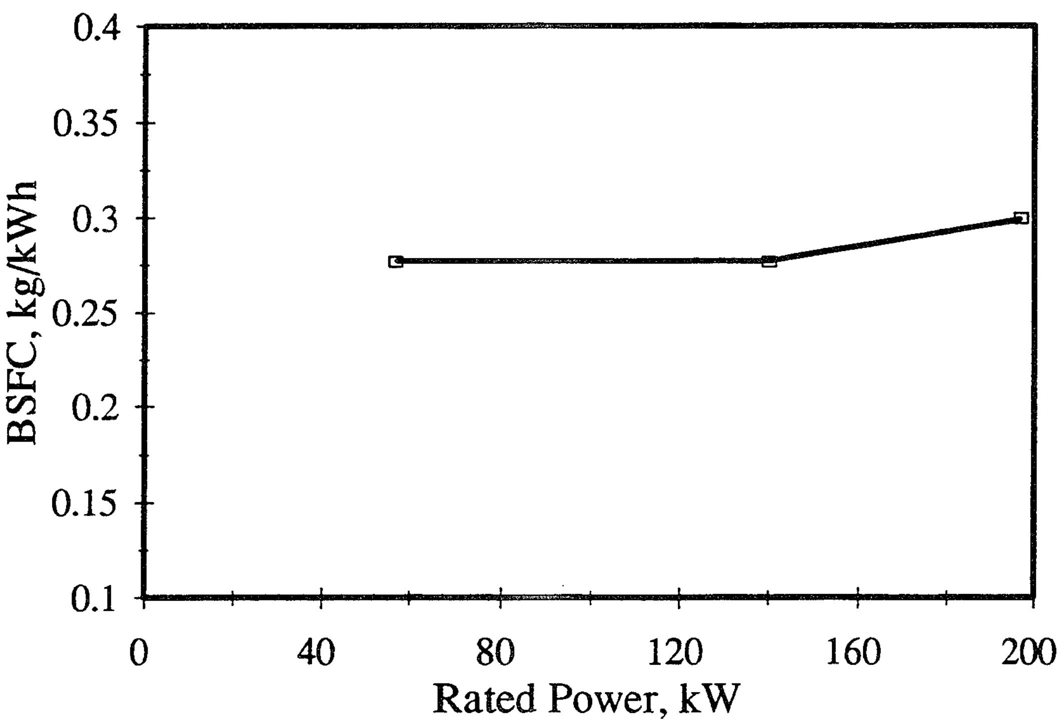


Figure 8. Fuel consumption of four-stroke spark-ignited engines

and NO_x is controlled by a combination of three-way catalyst and lean burn for lower in-cylinder temperatures. However, this technology is going the wrong direction for hybrid APUs, as it reduces power density.

A key consideration in emissions of gasoline engines for hybrid APUs, as it is for conventional vehicle applications, is the catalyst warm-up period. At start-up, the catalyst is cold and ineffective, and warm-up takes several minutes until the catalyst is fully effective. During this period, HC and CO emissions are high, both because of catalyst ineffectiveness and high engine-out emissions due to flame quenching on the cold cylinder internal surfaces and start-up enrichment of the fuel-air mixture. In conventional drivetrains, this happens only once per drive cycle. In hybrid applications, particularly for serial drivetrains, it may happen many times over the driving cycle. The problem will be highly sensitive to the control of the APU, in terms of the number of start-stop cycles encountered and the time between operating cycles during which the catalyst and engine cool down. An electrically heated catalyst can be used to overcome the problem of catalyst warm-up, at some added cost and penalty in power consumption. The APU engine application, then, will favor engines that have a short warm-up cycle and good combustion characteristics during cold start such that cold-start enrichment can be minimized. This will favor gaseous fuels over gasoline, and application of advanced techniques such as high in-cylinder turbulence, variable valve timing, and variable intake geometry.(8-10).

For the purpose of ranking, the four-stroke SI engines are assigned scores according to the above arguments for power density and fuel economy; on other criteria, the four-stroke SI engines are considered to be the baseline and are assigned a score of 1.0.

C. Four-Stroke Compression Ignition Engines

Diesel engines were largely discounted by Schneider and Burke because of a perception of low power density. This is a result of high cylinder pressures requiring heavier cylinder, piston and head structure, and lean-burn operation, which necessitates a higher air consumption than the comparative stoichiometric spark-ignition engines to achieve a target power level. With regard to technological development, these engines are highly developed for medium-duty and heavy-

duty highway vehicles, and also highly developed for stationary utility applications where weight is not a major issue. However, they are not highly developed in high power-density applications.

In terms of power density, the state-of-the-art is represented in Figs. 9 and 10. The state-of-the-art for specific weight (Fig. 9) is largely independent of power class and is, in fact, somewhat higher than that reported by Schneider (Fig. 1), based on data available to SwRI. A specific weight of 3 kg/kW would represent the best current technology for a 30-kW engine developed specifically for power density. Specific volume follows a similar trend (Fig. 10), with slight increase as rated power decreases, as would be expected. The state-of-the-art for a 30-kW engine is about 4 L/kW, higher than the values reported by Schneider of 3.1 for DI diesel engines and 3.7 for IDI engines. Since the rated speeds are limited to lower values by the combustion system, the generators would be expected to be proportionally heavier and bulkier.

The state-of-the-art for fuel consumption is shown in Fig. 11. This follows a gradually rising trend as power rating decreases. A level of about 0.23 kg/kW·h could be expected for a 30-kW engine; Burke reported levels of 0.2 for DI engines and 0.23 for IDI engines.

A word regarding direct injection versus indirect injection is appropriate. Direct injection is widely applied in larger displacement engines, and works in those applications primarily due to the ability to induce in-cylinder air motion by port design, and the ability to use heavy-duty fuel injection equipment with high injection pressures to achieve the required fuel-air mixing for good combustion. IDI systems are employed widely in smaller, higher speed engines where the rate of combustion must be enhanced by vigorous charge motion induced by the flow of air into a prechamber through a small orifice. This air motion enables the use of lower injection pressures and lighter injection equipment. IDI engines typically require higher overall compression ratios to overcome the fluid flow restriction across the prechamber orifice, but experience lower cylinder pressures due to the same restriction of flow as the combustion process proceeds. Fuel economy is degraded by the pumping losses into and out of the prechamber. IDI engines have been almost universally used in light-duty applications up until recent years. Advances in fuel injection and general engine technology, however, have now made DI systems more attractive

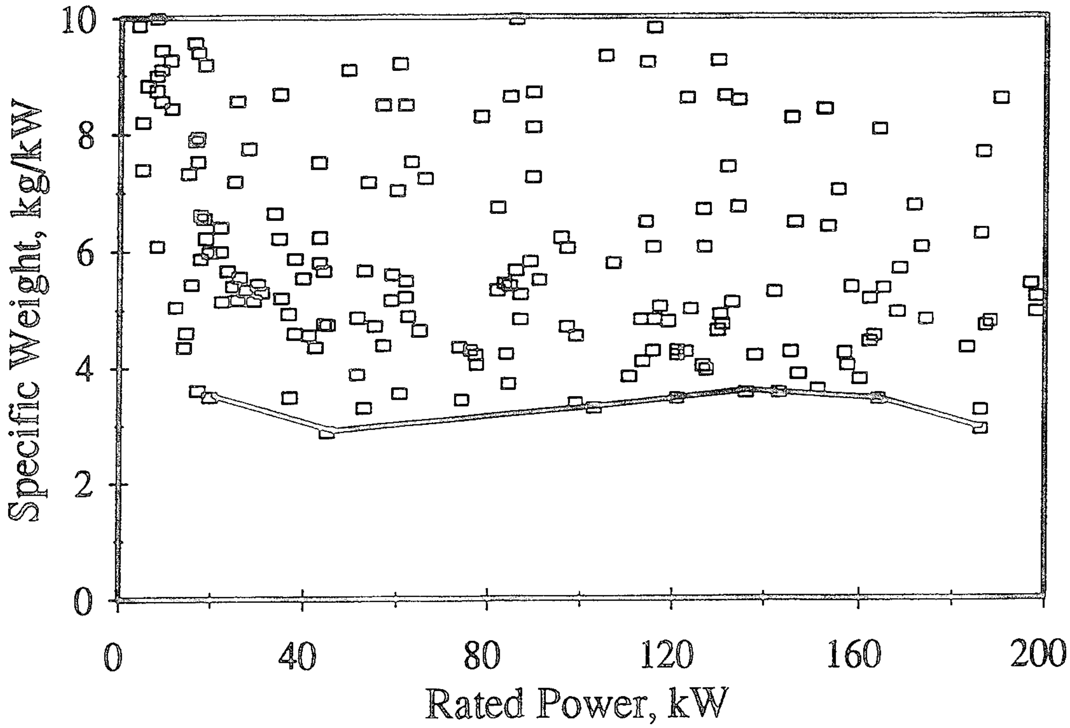


Figure 9. Specific weight of four-stroke diesel engines

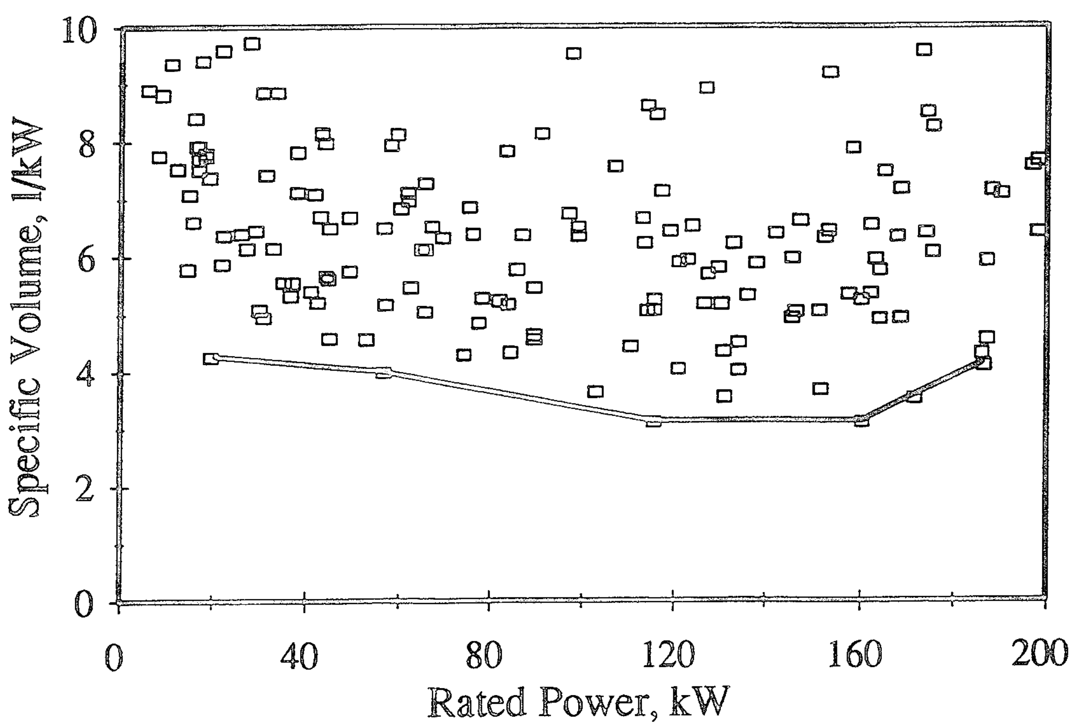


Figure 10. Specific volume of four-stroke diesel engines

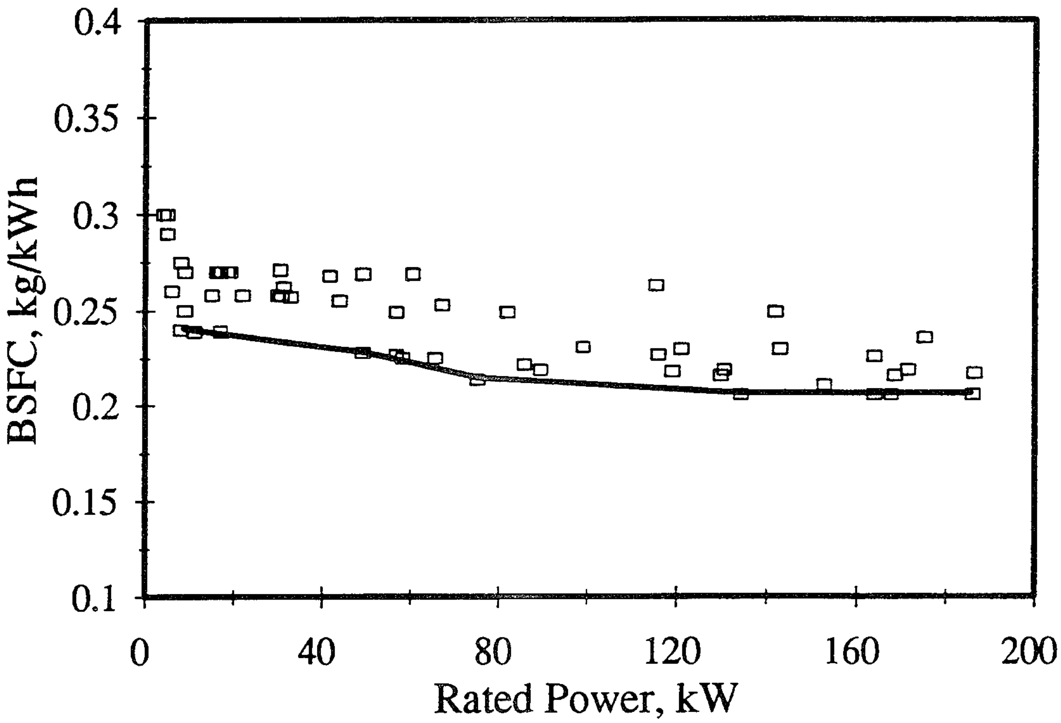


Figure 11. **Fuel consumption of four-stroke diesel engines**

to the automotive community. Particularly in Europe, the automotive DI diesel engine is making significant inroads. Audi produces perhaps the best engine in this class.(11) There is no fundamental reason why this technology cannot be applied to smaller output engines.

Emissions from diesel engines behave quite differently from those of spark ignition engines. The key emissions are particulates and NO_x. Substantial amounts of hydrocarbons can also be emitted, particularly during cold start. Aftertreatment for oxidation control of particulates and hydrocarbons is being introduced for many 1994 engines, but it is not as effective as aftertreatment for SI engines. Aftertreatment for NO_x is not yet feasible. Particulates are primarily produced when the engine undergoes hard transient load application or operates at high torque and low speed. NO_x primarily results from high load operation where the in-cylinder temperatures are highest. These engines offer the potential to be very low emitters when properly warmed up and when operated at constant speed and load outside the regions of high NO_x emissions; however, their power density at these conditions is not very good. The baseline emissions score assigned to diesel engines is 0.9.

The cost of current diesel engines is somewhat higher than that for SI engines as a result of more sophisticated fuel injection equipment and generally robust construction. However, higher technology SI engines are comparable in cost and may be more expensive and heavier if designed to equivalent reliability, which is excellent for diesel engines. Producibility is comparable to four-stroke SI engines. Transient response is slightly worse than that for SI engines because of larger reciprocating and rotating mass.

Cranking torque is an issue with diesel engines because of the high compression needed to attain the temperatures to sustain combustion. Smaller engines are more of a problem in this area because of torque fluctuations with fewer cylinders. Cold startability of smaller engines is also a concern because of higher relative heat transfer from the combustion chamber. This may become an issue with APUs in frequent start-stop operation. However, with appropriately designed power electronics and controls, the generator can also be used as a starter motor, providing high cranking torque to overcome these issues.

NVH problems are significant for diesel engines, primarily due to the high rate of pressure rise in the initial stages of combustion. Automotive diesel engines are typically associated with a "clatter" noise, the sound of diesel combustion. This noise can be regulated by injection rate control and by turbocharging, at significant increase in cost. Turbocharging may not be an option for a 30-kW engine.

Multifuel capability is less with diesel engines than with SI engines when considering light hydrocarbons. Diesel combustion will only work with fuels that contain significant quantities of long-chain hydrocarbons, which will initiate the combustion process by thermal decomposition to shorter molecules and free radicals. Organic fuels such as vegetable oils are potential alternatives, but natural gas, propane, hydrogen, or other alternatives cannot be used without significant changes to the combustion system.

D. Two-Stroke Spark-Ignited Engines

The top ranking in the analyses by Schneider and Burke for two-stroke SI engines was based on the status of research at that time, indicating great promise for automotive two-strokes with direct fuel injection. Eleven years later, these engines have not yet lived up to their predictions of rapid inroads into automotive application. The primary reasons for this slower-than-expected development are difficulties with durability and emissions. The durability issues are associated with lubrication of the piston-ring-cylinder wall interface, particularly where the rings are used to control ports in the cylinder walls for admission of air and exhaust of products. Durability of crankshaft parts is also an issue in those engines that utilize crankcase pumping as a means of supplying scavenging air and cannot use conventional pressure lubrication and plain bearings. Catalyst durability is a problem because of the presence of lubricating oil in the exhaust. The emissions issues are related to the fact that two-stroke engines inherently contain excess oxygen in the exhaust, which makes the application of three-way catalysts impossible. Unburned hydrocarbons and CO can be effectively treated with oxidation catalysts, but research has yet to perfect a lean catalyst for NO_x emissions. Nevertheless, research continues into automotive two-stroke engines, and reports from key developers such as Orbital Engine Company and Chrysler suggest that these problems are not insurmountable.

The key advantages of the two-stroke engine are small, lightweight construction (high power density, both volumetric and weight) and low cost associated with smaller parts count. The elimination of the overhead valvetrain is largely responsible for both of these advantages; yet some researchers are still applying overhead valves in their two-stroke engines to overcome the durability and exhaust oil contamination issues.(12-15)

The availability of specific engine data for automotive class two-stroke engines is poor because of the highly competitive environment; current developers are not anxious to release their data. Thermal efficiency gains attributed to lower friction are generally offset by losses associated with scavenging and pumping; thus, thermal efficiency of the two-stroke engines is comparable to that of four-stroke engines. Information obtained in an industry survey by SwRI in 1991 indicates that automotive class engines can achieve a specific weight of 0.9 kg/kW (16); this level is also

probably reasonable for APUs. Despite higher "weight overhead" associated with smaller size, these engines would also benefit from power density associated with higher speed. No data was obtained on specific volume, but a level of 0.7 times that for four-stroke SI engines is reasonable, accounting for the reduction in engine height associated with elimination of the overhead valves, and for the reduced size and weight of a generator running at higher speed.

Cost of piston-ported two-stroke engines is significantly lower than the poppet-valved engines because of the greater simplicity and lower parts count. The same argument holds for gains in producibility and reliability. Emissions are harder to control. The technical risk is higher because of the immaturity of this technology. NVH is probably about the same as with four-stroke engines. Mechanical noise may be lower without the valvetrain, but exhaust noise and bearing noise are increased for two-strokes with rolling-element bearings. Transient response should be better than for a four-stroke engine because of reduced overall inertia. Cranking torque fluctuations for a multicylinder engine should be lower because there is a compression stroke on each revolution.

E. Two-Stroke Compression Ignition Engines

Two-stroke diesel engines are in widespread use for high horsepower applications such as locomotives and ships. The scaling of two-stroke compression ignition technology to small, lightweight engines has been largely ignored due to lack of application. However, the military interest in small power plants for unmanned aerial vehicles (UAVs) that burn diesel or jet fuels has renewed the interest in small two-stroke diesels. SwRI has developed a design for a UAV engine that is of appropriate size and possesses the characteristics for the hybrid APU.(17) This design was demonstrated in limited testing, and further development is needed to take it to production-ready status. Key features of the engine are piston compressors driving scavenging air, allowing for cooling of the backside of the power pistons by oil jets; reed valve controlled transfer ports; and an IDI combustion system using low pressure injectors, unit injector pumps, and air cooling. The design targets are 30-hp (22-kW) output at 4,500 rpm, with BSFC of 0.25 kg/kW·h. Production weight was estimated at 35 lb (16 kg). Operating speed is somewhat lower than for SI combustion systems, resulting in greater generator size and weight. This engine

will be the benchmark for comparison. For greater reliability, some weight would be expected to be added to this engine design, such that estimated engine weight for a hybrid APU would be around 22 kg. Because of the high compression ratio requirements and the heat losses working against compression ignition, cranking torque and cold startability will be somewhat worse than with spark-ignited engines.

F. Wankel Rotary Engine

The Wankel rotary engine, invented by Felix Wankel in the mid-1960s, is certainly the most well-developed of rotary engine concepts, having been in production in automotive and aerospace applications since the early 1970s. The development status of this concept is such that the technical risks associated with its application are well known and have been dealt with in great detail. Its key advantage over state-of-the-art reciprocating engines is power density on both weight and volume basis; with direct injection, its fuel consumption can also be competitive.

The basic principle of operation of the Wankel engine is shown in Fig. 12 (reproduced from Reference 18). A three-cornered rotor is constrained to rotate about its center of gravity, which in turn orbits around the crankshaft centerline. The rotor turns at one-third the crankshaft speed, driven by a gear on the crankshaft meshing with a ring gear on the rotor. The three corners divide the trochoid-shaped rotor housing into three working chambers, each of which executes a full four-stroke cycle on each rotation of the rotor. Thus, the engine achieves one power stroke per revolution of the crankshaft. By counterbalancing the crankshaft, the system is completely dynamically balanced.

The limitations of the system arise from several sources. There is a large amount of surface area compared with the chamber volume; therefore, heat transfer rates are high, causing relatively higher heat rejection and associated inefficiency and component loads. Also, combustion quenching is high, resulting in more difficulty in controlling hydrocarbon and CO emissions from their source in a homogeneous charge. These problems have led most advanced engines to use stratified-charge combustion systems, confining the fuel-laden charge to a portion of the useful

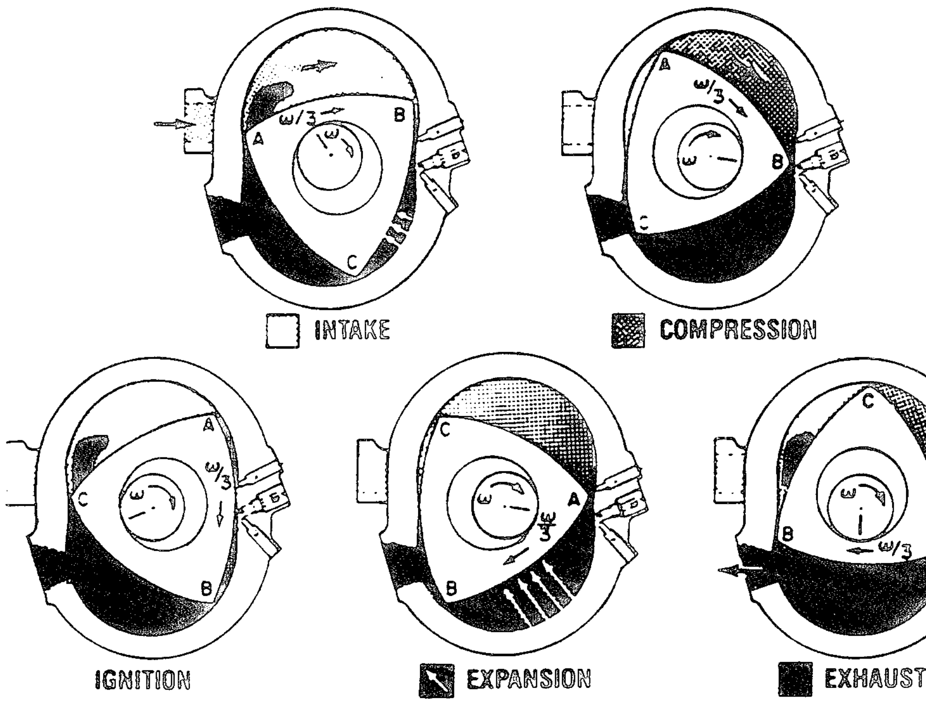


Figure 12. Basic operation of the Wankel rotary engine

volume, which reduces the power density. The high heat transfer rates cause cooling problems in the rotor, since a large portion of its surface is exposed to hot gases. Internal cooling by oil is the most common method of rotor cooling; air-cooled engines have great difficulty in this area. Sealing of the rotor apex and sides has been the focus of much development, and problems in these areas are generally solved; however, the large amount of rubbing contact, along with fairly high operating speeds, leads to higher friction in comparison with piston engines.

Despite these issues, the Wankel rotary engine has experienced some degree of success in research applications where high power density is a priority, and still retains interest among developers of lightweight aircraft, particularly unmanned aerial vehicles. Significant development activities continue for commercial and military applications at Mazda Motor Corporation (19, 20), John Deere Technologies International, Inc. (now Rotary Power International) (18, 21, 22), and AAI Corporation (23). Also, a single-rotor air-cooled engine of 38-hp output is manufactured for UAV and target drone applications by Alvis UAV Engines Ltd.(24) The available data regarding Wankel engine power density and fuel economy are shown in Figs. 13 through 15. From the limited data available, it is clear that these engines significantly challenge the power

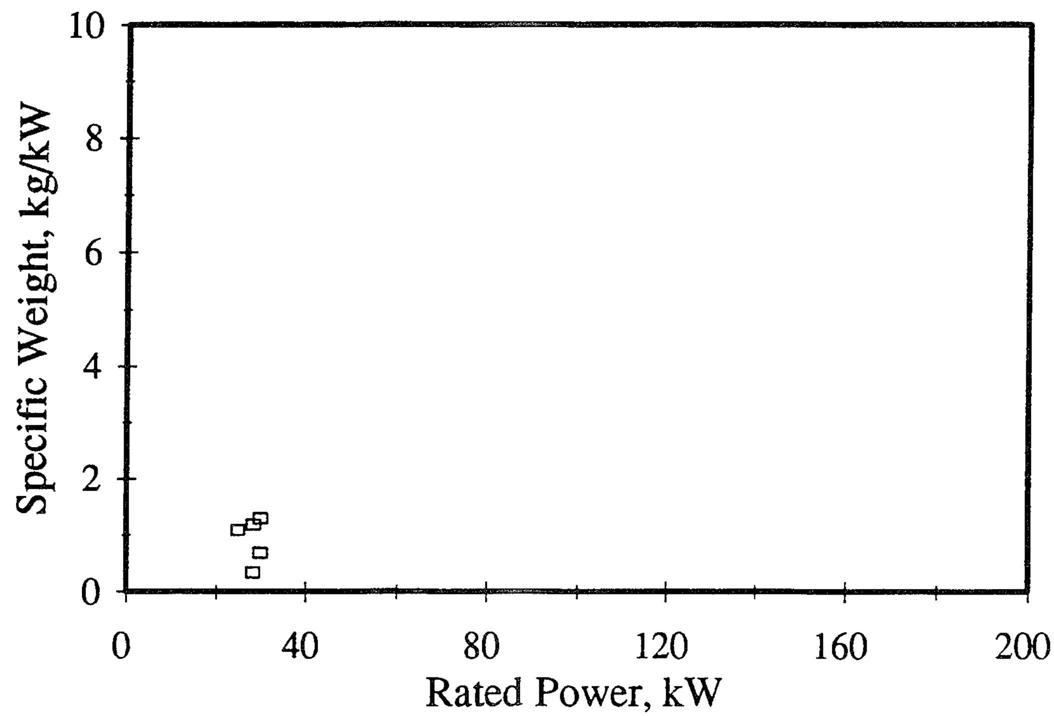


Figure 13. <u>Specific weight of Wankel rotary engines</u>

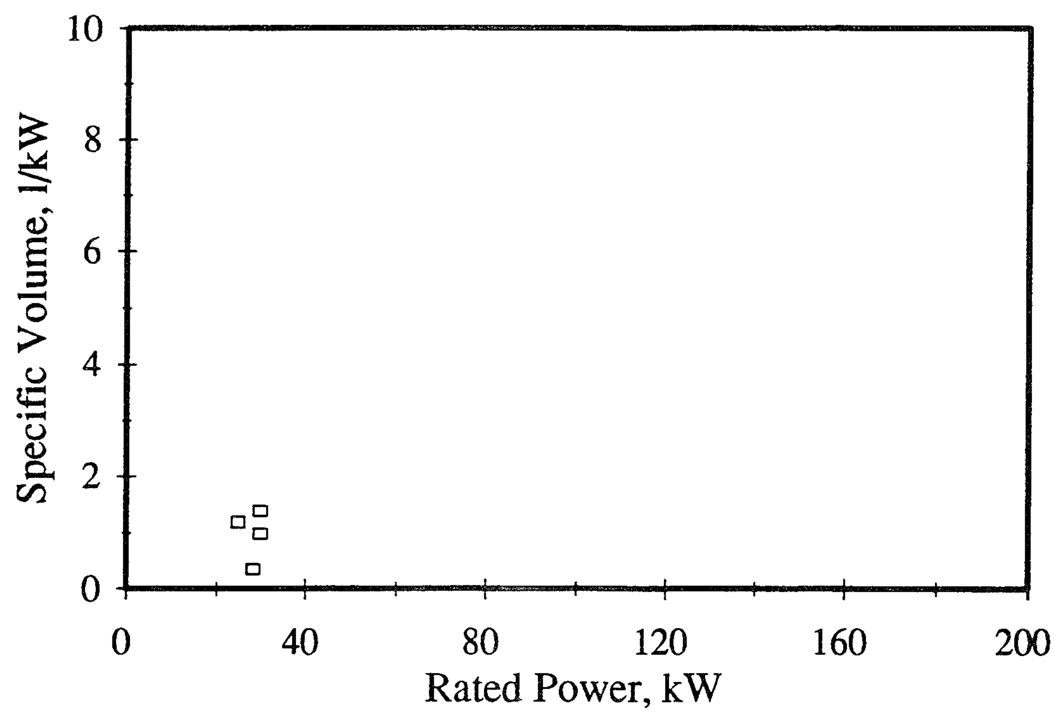


Figure 14. <u>Specific volume of Wankel rotary engines</u>

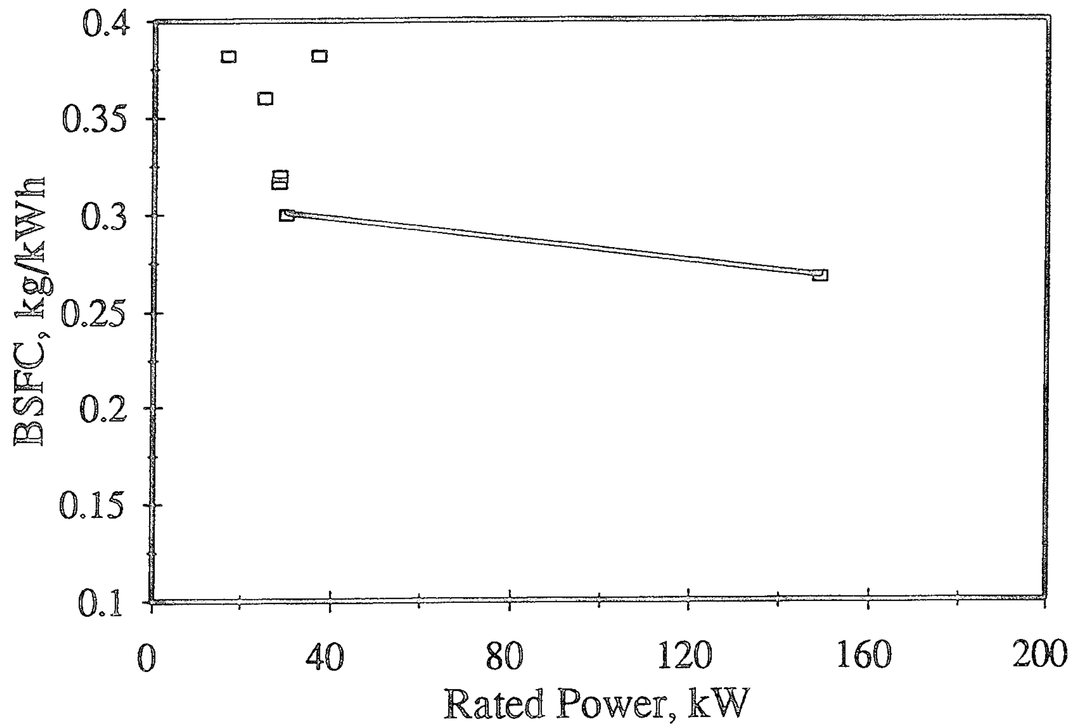


Figure 15. Fuel consumption of Wankel rotary engines

density of conventional reciprocating engines in the under 40-kW power class and can nearly compete with conventional technology in fuel consumption.

For an engine developed for the APU system, thermal efficiency could be expected to be 27 percent; coupled with the generator, overall efficiency would be 25 percent. Engine specific weight and volume of about 0.7 kg/kW and 0.6 L/kW, respectively, appear feasible. Operating speed could be at about 8,000 rpm, resulting in generator specific weight and volume of 1.18 kg/kW and 0.26 L/kW, respectively.

Cost of the Wankel engine should be close to that of conventional two-stroke engines due to the simplicity of the mechanism. Emissions may be more difficult to control. Producibility and reliability should be good. Since there is no reciprocating motion, the sliding surfaces never reverse their sliding motion, which is a prime cause of wear in reciprocating engines. Also, Wankel engines use total consumption lubrication, which means that fresh oil is continually supplied to the rubbing surfaces and cannot become degraded or diluted by mixing with fuel. The technical risks are well established but somewhat higher than for conventional four-stroke engines. Noise and vibration are well controlled by the balanced system. Multifuel capability

is questionable; however, rotary engines have run successfully on gasoline, heavy fuels, and gaseous fuels. Transient response should be equivalent to two-stroke engines. Cold startability should be good, with smooth and low cranking torque.

VI. ADVANCED CONCEPTS

The advanced concepts considered in this study are explored in the sections that follow. The concepts are classified either as "APU System Concepts," which involve a radical departure from conventional engine technology, and "Technology Concepts," which include ideas that can be integrated with otherwise conventional engine systems. The system concepts are scored and ranked along with the conventional technologies; technology concepts are scored and ranked separately.

A. APU System Concepts

1. Free-Piston Engine

The free-piston engine (FPE) is an idea that has been considered by many researchers since the early 1900s. The basis of the concept is a piston reciprocating in a cylinder without any kinematic constraint or mechanical connection, as illustrated in Fig. 16. Power is produced by a more or less conventional combustion system operating on a two-stroke cycle, firing once per engine cycle. The combustion system can be diesel, spark-ignited, or can consider some of the advanced concepts discussed later. The thermal energy is converted into kinetic energy in the piston and is then absorbed by one of several means and turned into useful work. As the kinetic energy is absorbed, the piston slows and reverses direction, returning to the combustion chamber for the next stroke. The reversal is aided by pressure building up in a chamber at the opposite end of the device from the combustion chamber, typically referred to as the "bounce" chamber. The precise means of controlling the piston motion without hitting either end of the cylinder is the subject of much research and has been successfully demonstrated in several cases.

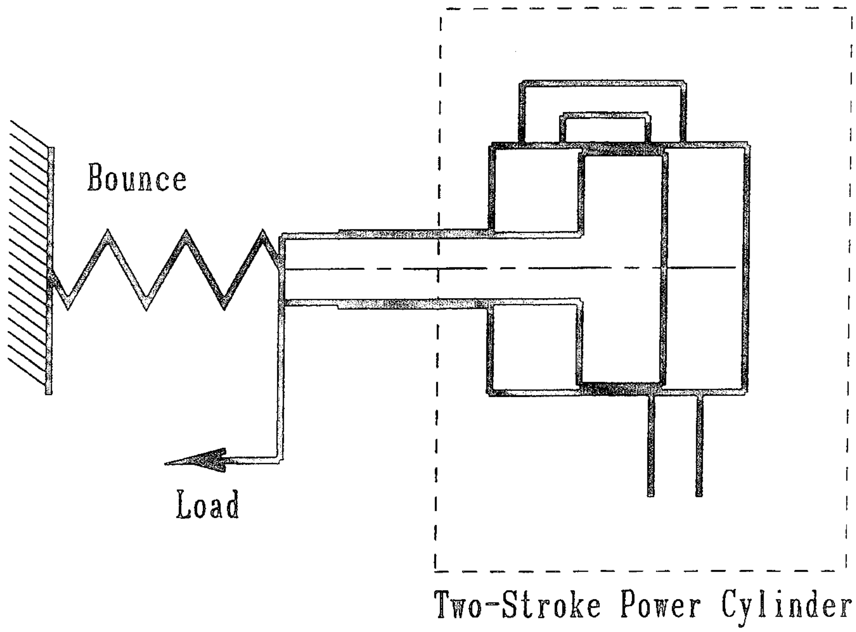


Figure 16. Basic concept of free-piston engine

The FPE has several peculiar characteristics that set it apart from conventional engines. It operates at theoretically higher mechanical efficiency due to the reduced friction from the fewer number of moving joints and mechanical parts. It operates over a very narrow reciprocating speed band because it is controlled by the natural frequency of the spring mass system, the mass being the piston and the spring being the effective gas spring of the compression and expansion events at either end of the piston. The speed can be controlled to a certain extent by modifying the compression ratio and the gas pressures. The compression ratio and piston stroke are variable and can be varied stroke-by-stroke. However, the stroke range is limited by the natural dynamics and cannot be controlled independently of operating conditions. Its transient response is very good, as it reaches its operating speed essentially instantaneously. One drawback is that because of the narrow speed band and the need to maintain appropriate compression ratios for the targeted combustion system, the ability to turn down from the rated condition is limited. The engine is ideally suited to fixed operating point cycles, as in the APU application.

The primary perceived advantages of the FPE over conventional technology are as follows:

- Thermal efficiency – Eliminating crankshaft friction should result in a thermal efficiency improvement of 3 to 5 percent;

- Cost and reliability – Fewer moving parts means less cost and more reliability;

- Mechanical integrity – Combustion pressures are not limited by structural considerations in the crank system;

- Transient response – Transient response is very good;

- Power density – The power density advantages, if any, are not as clear, as the engine cannot be run at high speed independently of the natural dynamics. High power density can be achieved by running at high boost levels, with higher natural frequency owing to the increased gas spring constant.

The earliest recorded work with the FPE concept was by the Marquis R. De Pescara, an Argentinean researcher working in France who patented the concept in the U.S. in 1928.(25) His patent covered a gas generator application, whereby the power from a diesel cylinder was absorbed by an air compressor which drove the scavenging air through the power cylinder into a gas turbine. All shaft power output was provided by the gas turbine. This concept was developed commercially by the Als-Thom Company into a 770-hp unit, and later by the SIGMA organization in France, resulting in the GS-34 engine rated at 1,200 hp. The GS-34 unit was installed on several ships and in stationary power plants and saw considerable service in the 1930s.

After WWII, sporadic development of free-piston gas generators was pursued by researchers in France and England.(26) General Motors (GM) Corporation and Ford Motor Corporation both pursued free-piston engines as a vehicle power source in the 1950s.(27-29) Both companies focused on the gasifier-turbine configuration, wherein the FPE produces no output work other than the high-pressure exhaust gas, which is routed to a turbine to produce shaft power. GM took the work as far as a vehicle demonstration. Their device was a siamesed pair of inward-

compressing diesel cylinders with stepped bores such that the compressor cylinders were of larger diameter than the engine cylinders. This is shown conceptually in Fig. 17.

Each pair of power pistons was synchronized by a mechanical linkage, but there was no linkage between the two siamesed units to maintain their phase relationship. Instead, pneumatic means were used to maintain their 180° out-of-phase operation, thereby permitting optimum utilization of the compression pulses from one unit to scavenge the other unit. This demonstrates that phase control of FPEs is feasible by other than mechanical means. The engine in fact started in parallel operation but achieved its proper phase relationship within a few strokes. Starting was by pneumatic means with an accumulator supply pressure of 206 kPa (30 psi).

For tractive power, the GM 4-4 Hyprex engine used a five-stage axial turbine, with a 7:1 gear reduction into a four-speed transmission. The peak turbine efficiency was approximately 70 percent. Fuel efficiency of the FPE was stated in terms of "gas hp-hr," being based on the exhaust stream pressure and temperature rather than shaft power output from the turbine. Demonstrated fuel consumption was 0.45 lb/gas hp-hr (0.274 kg/kW·h), which translates to a gas thermal efficiency of approximately 30 percent. This is much lower than might be hoped for from this device. With the turbine efficiency, the brake thermal efficiency to shaft output would be about 21 percent. The researchers predicted a developed fuel economy of 0.36 lb/gas hp-hr (0.219 kg/kW·h), which with a well-developed turbine of 80 percent efficiency would result in thermal efficiency of 30 percent, still much lower than current crankshaft engine technology. One reason for this less than stellar performance was pointed out by Mr. Samolewicz: the free-piston gasifier must move substantially more air than it utilizes in the combustion process.(26) Irreversible energy losses are associated with this air handling and pumping. The large volume of air consumption must also be accommodated by larger filters and ducting than would be required for a conventional engine of comparable power output, which is a space claim disadvantage.

Specific weight demonstrated in these experiments of the 1950s was near 3 lb/hp (1.8 kg/kW). Specific weight could be reduced by lighter materials and by increasing the cyclic speed, which

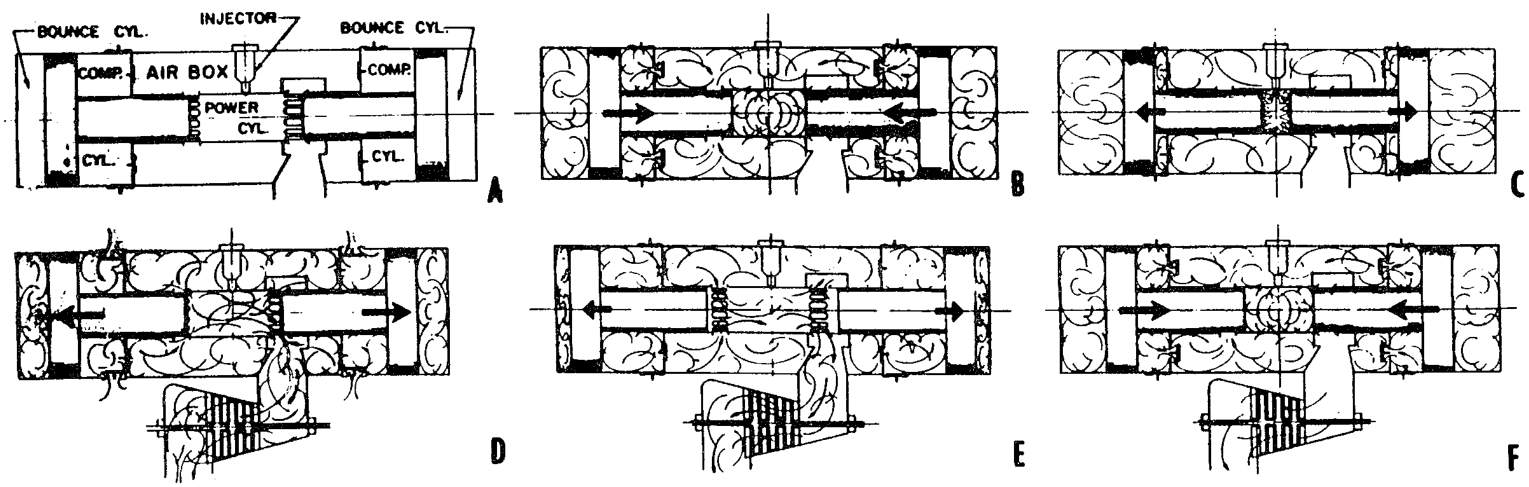


Figure 17. Operating principle of GMR 4-4 Hyprex free-piston gas generator

can be accomplished by greater boost, higher bounce pressures, and reduced reciprocating mass. All of these are development challenges.

In the mid-1960s, free-piston gasifier research was carried on in Canada by the Free-Piston Development Company and at the National Research Council of Canada.(26, 30) The unit developed was considerably smaller than those of previous research, at about 65 gas hp (48 kW), approaching appropriate size for hybrid APUs. Its weight and volume were 410 lb (186 kg) and 4.69 cu. ft. (133 L), respectively, without accessories. Without provision for a power turbine and alternator, but assuming a gas-to-electric conversion efficiency of 75 percent, the specific weight and volume work out to 5.1 kg/kW and 2.7 L/kW, respectively. Minimum specific fuel consumption was approximately 0.45 lb/gas hp-hr, consistent with that developed by the GM researchers. These numbers are not encouraging. The reciprocating speed was 2,015 cpm. Further development of the system for higher speed and lighter weight could presumably improve these numbers somewhat. Predictions by Wallace, et. al. (30) were that at high boost pressures and turbine pressure ratio of 5 to 1, the FPE gas generator-turbine combination could achieve overall brake thermal efficiency of 39.2 percent, still short of current diesel engines but competitive with spark-ignition engines. At the higher cycle pressures and temperatures, NO_x emissions may become an issue.

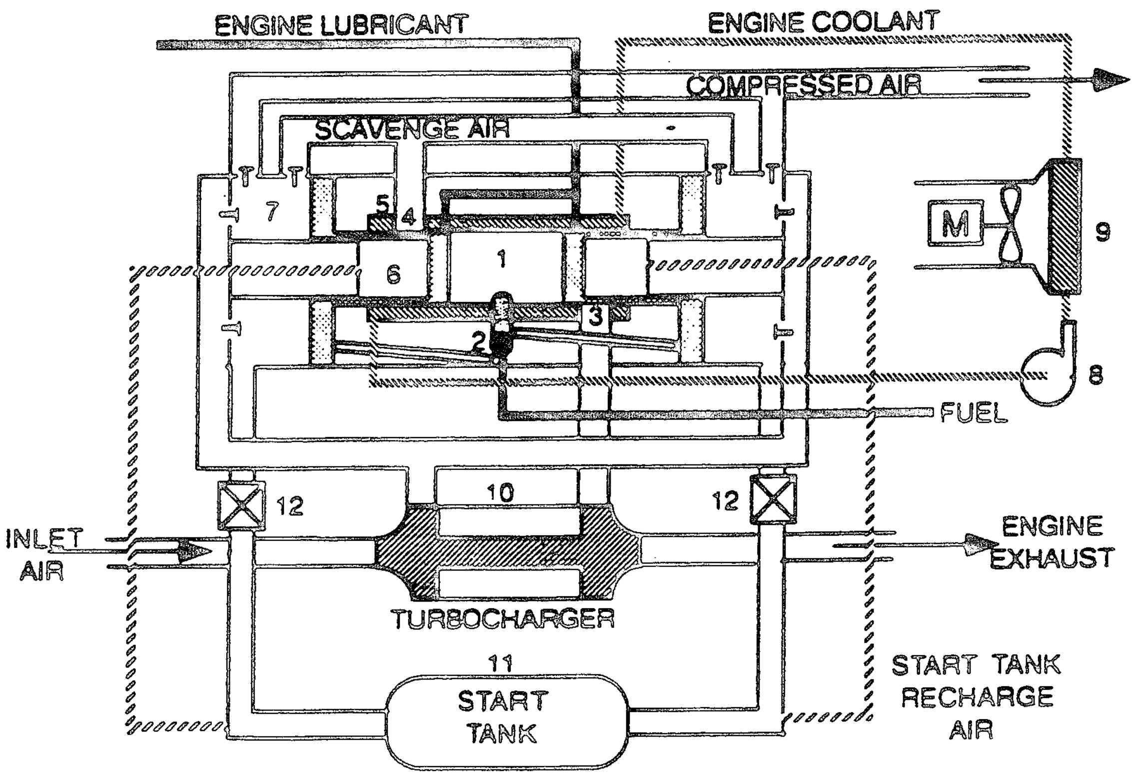
More recent development of FPEs has focused on their use as dedicated gas compressors and hydraulic pumps. As a hydraulic pump, the FPE could be used in hybrid vehicles in one of two ways: hydraulic motors could be used for traction or for driving an electric generator. Since this study focuses on the hybrid electric vehicle, the traction motor configuration will not be pursued here but is well worth considering in the overall view of high efficiency vehicles. Under certain conditions, hydraulic motors can have efficiencies on the order of 95 percent or better, well in excess of that of state-of-the-art turbine expanders; therefore, the FPE-hydraulic pump/motor/generator combination is an attractive alternative to the FPE gas generator.

Research in Japan by Dr. A. Hibi of Toyohashi University has demonstrated several aspects of the FPE hydraulic pump.(31, 32) Their solution to the problem of poor turndown is to intermittently cycle the engine, with a dead period between cycles. In demonstration testing, the

reported overall thermal efficiency of their device is poor at 14.3 percent; however, they demonstrated good conversion of indicated gas power to hydraulic power at an efficiency of 84 percent, including losses to drive a scavenging pump. The poor thermal efficiency can be attributed to the use of a motorcycle-type carbureted two-stroke cycle, with inherent losses of unburned fuel-air mixture during scavenging. In a later publication, Dr. Hibi suggested an attainable overall thermal efficiency of 41 percent; however, his assumption of 50 percent indicated thermal efficiency for the combustion process is optimistic.(33)

The FPE hydraulic pump, with a spark-ignited combustion system, was studied analytically by P.C. Baruah.(34) A fairly comprehensive first-principles thermodynamic simulation was used, including the effects of flame speed on combustion rates. However, the analysis omitted the gas exchange effects, which may be significant. In comparison with a conventional crank engine, this study pointed out one reason why the thermal efficiency of the FPE may not reach its full perceived potential. The reversal of the piston at top dead center is controlled entirely by the dynamics of the spring mass system. Upon commencement of combustion, the piston undergoes rapid acceleration toward the load end of the engine and achieves substantially higher velocities during the expansion stroke. The timing of combustion is less flexible than with conventional engines and must take into consideration the need to appropriately control compression ratio. As a consequence, the combustion process tends to occur over a greater proportion of the expansion stroke, in terms of volume, not time. Thus, the thermal energy of combustion is used less effectively. To overcome this problem, FPEs would have to achieve higher rates of combustion. One advantage of later combustion, as pointed out by Mr. Baruah, is lower NO_x emissions.

In gas compressor applications, FPE research has been carried out recently by AiResearch Los Angeles Division of Allied Signal Aerospace Company (35) and by Tectonics, Inc. (36). The AiResearch work focused on a compressed air supply for Army tanks and developed an FPE design similar in concept to those developed by GM and Ford. Since clean, compressed air was the desired output, an outward-compressing arrangement was used, with two opposed pistons and a diesel combustion system, as shown in Fig. 18. This arrangement allowed the use of the larger



1. Diesel cylinder
2. Fuel injector and synchronizing mechanism
3. Diesel cylinder exhaust port
4. Diesel cylinder scavenge port
5. Diesel cylinder cooling jacket
6. Bounce cylinder
7. Compressor cylinder with reed valves
8. Coolant pump
9. Coolant radiator and cooling air fan
10. Turbocharger
11. Start tank
12. Start valves

Figure 18. Conceptual design of AiResearch Mark II free-piston compressor

bore to supply air both for scavenging and for the output. The best obtained indicated specific fuel consumption of this unit was 0.388 lb/hp-hr (0.236 kg/kW·h), based on the power cylinder indicator diagram. However, based on the compressed air output, specific fuel consumption was 0.621 lb/hp-hr (0.378 kg/kW·h), representing thermal efficiency of approximately 22 percent. About half the difference between "indicated" and "brake" output can be attributed to mechanical friction and half to the portion of the compressed airstream diverted to scavenge the engine. Some of this pumping loss could presumably be averted by rearranging the valving and porting such that the scavenging air is diverted at the desired pressure, not first pumped to the compressor output pressure. Thermal efficiency of 25 to 28 percent might be attainable by using this arrangement.

Tectonics, Inc. was engaged in the development of FPEs for dedicated air and gas compressors before the company dissolved in 1993. SwRI was contracted with Tectonics for the design and development of their FPE compressors. The units designed by Tectonics were targeted for stationary applications and were not designed for compactness or weight. A single, conventionally scavenged two-stroke SI natural gas engine cylinder was used for the power unit, driving a conventional single- or multistage compressor cylinder with a rack-and-pinion-driven counterweight for balance. In tests at SwRI laboratories, these units achieved about 23 to 24 percent gas thermal efficiency. Inefficiencies were attributed to incomplete combustion, scavenging losses, heat transfer, and friction primarily in the ringpacks, which were not optimally designed.

The key to achieving high power density in FPEs is high cyclic speed. It has been reported that Mr. Frank Stelzer of Germany has designed a free-piston engine capable of cyclic speeds on the order of 30,000 cpm.(37, 38) This is achieved by employing two opposed combustion chambers driving both sides of the piston, such that the bounce pressure is the combustion pressure. The claim of 30,000 cpm seems doubtful, but it is likely that cyclic speeds could be boosted substantially by this approach. The limit to reciprocating speed will most likely be ring wear.

There are several possibilities for applying FPE technology to a hybrid APU. These are discussed as separate engine concepts.

a. *Free-Piston Engine With Linear Generator (FPELG)*

An approach to power production is to provide a linear electric generator at the "bounce" end of the cylinder. The concept is shown schematically in Fig. 19. This generator will convert the kinetic energy of the piston directly to electric current, slowing the piston motion in the process. SwRI, in partnership with The University of Texas Center for Electromechanics (UTCEM) and sponsored by ARPA, has investigated the possible design configurations for a linear alternator and developed a design concept that integrates nicely with the FPE and has high predicted efficiencies.(39) The elements of the generator include an iron core with an air gap, an excitation coil, a generator coil, and the lower edge of the piston skirt. Magnetic flux is induced in the iron core by the excitation coil. As the piston skirt passes through the air gap, it alters the shape of the magnetic field, inducing current in the generator coil. This concept is mechanically simple in that the only moving part is the piston itself, and there is no requirement for electrical contact to the piston. For this reason, mechanical losses should be extremely low. The nature of the device is such that current is only generated during the expansion stroke of the cycle, and the power generation during expansion also exerts a restoring force on the piston, helping to slow its travel toward the bottom end. Gas bounce is necessary to achieve the next compression stroke. A test is currently underway at SwRI to demonstrate the highest risk element of this concept, the generator.

Preliminary design of the SwRI-UTCEM FPELG focused on a demonstrator unit of approximately 6-in. bore and stroke, based on the Tectonics engine unit. Design calculations indicated that this concept could achieve a peak thermal efficiency of approximately 30 percent, including generator losses. The demonstrator unit was not packaged for minimum weight; however, power-to-weight ratios of about 2.5 kg/kW are probably achievable. The demonstrator represents a specific volume of about 2.8 L/kW, and with development, the level of 1.8 L/kW is probably achievable, including power electronics.

Cost, reliability, and producibility should be much better than with conventional engine-generator configurations because of the tight integration of the system and the single moving part.

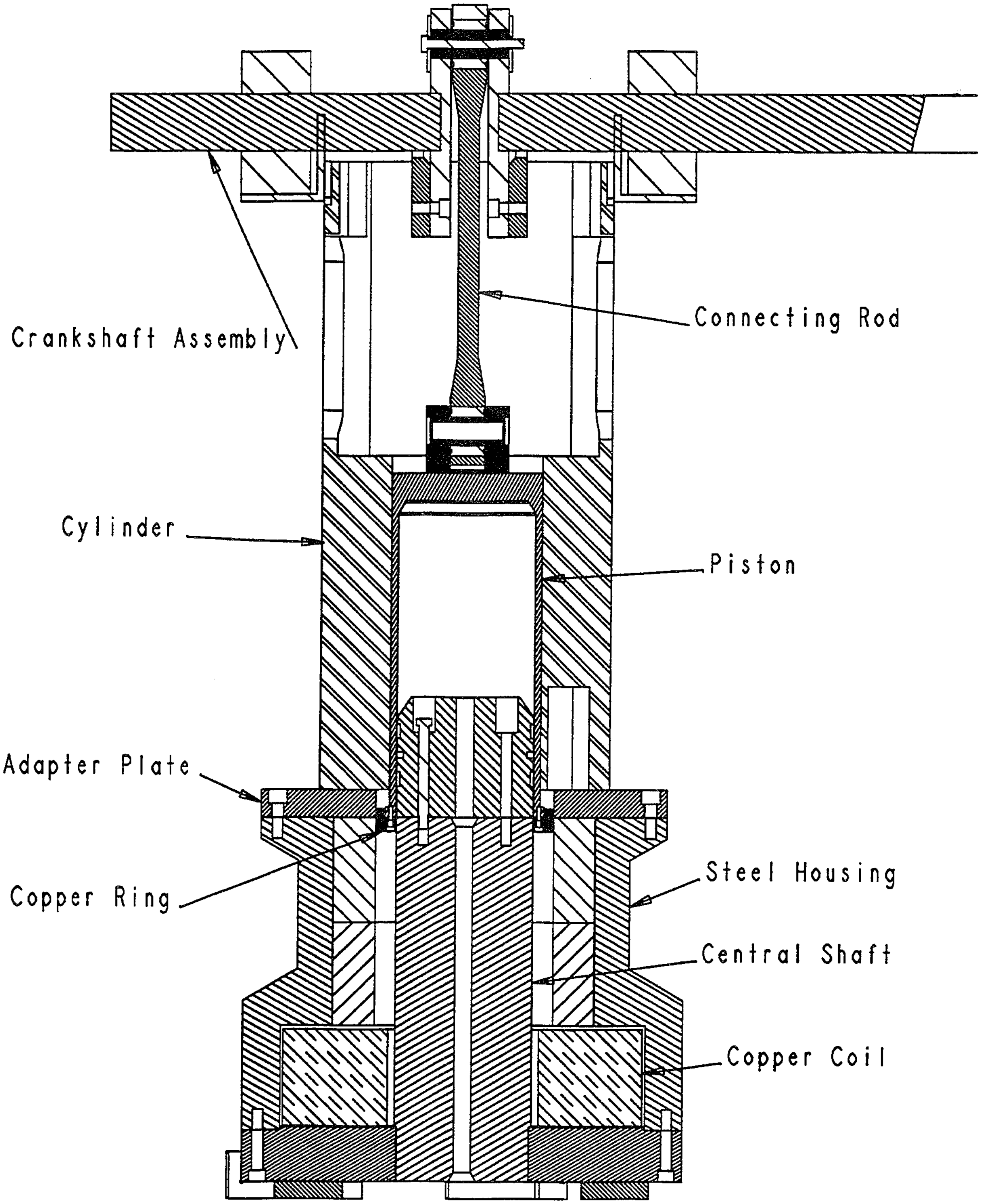


Figure 19. <u>**Linear generator for free-piston engine in rig testing configuration**</u>

Emissions issues are the same as with conventional two-stroke engines. There is great technical risk associated with this technology, as the generator concept is unproven and the integration of the engine with the generator presents challenges. Using two opposed units, there are no unbalanced reactions, so NVH should be better than with conventional engines; however, exhaust and intake pulsations will have to be dealt with. Multifuel capability is a bonus for the FPE, since its compression ratio can be adjusted to suit the fuel used. Transient response is excellent, as the engine reaches its operating speed instantaneously. This is also reflected in startability; for a properly designed system, starting is accomplished with a single impulse from either the electromagnetic system of the generator or a compressed air bottle attached to the bounce chamber.

b. Free-Piston Gas Generator With Turboalternator

Another option for using the FPE technology is to update the technology of the 1950s and 1960s by routing the exhaust gas to a turboalternator. The FPE technology for gas generators is fairly well developed, if somewhat obsolete. Updates of this technology could be expected to achieve overall specific fuel consumption of 0.48 kg/kW·h, including 0.35 kg/gas kW·h for the gas generator, turbine efficiency of 80 percent, and generator efficiency of 92 percent. This translates to a thermal efficiency of roughly 17 percent. Gains would be made in specific power and weight owing to the high speed generator. Gas generator specific volume could be around 2.0 L/kW and specific weight about 1.5 kg/kW, if developed specifically for these criteria.

Cost of the turboalternator may be somewhat higher than the cost of the linear generator plus power electronics required in the previous concept; in addition, the FPE is somewhat larger and more expensive because a stepped bore and valving are needed for the compressor cylinder. Emissions again are equivalent to the two-stroke engine. Producibility is probably equivalent to the conventional four-stroke engine, gains of the FPE compensated by complexity of the turboalternator. Reliability should be somewhat better than conventional engines, as there are fewer moving parts, and the turbine does not see high exhaust temperatures. The technical risk is relatively low, as each component has been fairly well developed. With a balanced FPE, the noise and vibration characteristics should be even better than the FPELG because the turbine will

serve to muffle exhaust noise. Multifuel capability is the equivalent of the FPELG. Transient response is not as good as with the FPELG, or even with conventional engines, because of the lag of the turboalternator. However, startability is as good, relying upon a compressed air supply. The compressed air can come from a bottle and can be stored during system operation.

2. Rotating Combustion Chamber Engine

In 1993, SwRI contracted with Mr. Ross Riney to evaluate his concept of a rotary engine.(40) The essence of the concept includes separate compression and expansion rotors on a common crankshaft, with a combustion chamber attached to a third rotor on a shaft rotating at one-half crank speed at right angles to the crankshaft. This is illustrated in Fig. 20. Floating vanes riding on the rotors divide the compression and expansion housings into two chambers, allowing for intake and compression processes to occur simultaneously in the compression rotor, and expansion and exhaust processes to occur simultaneously in the expansion rotor. The air charge

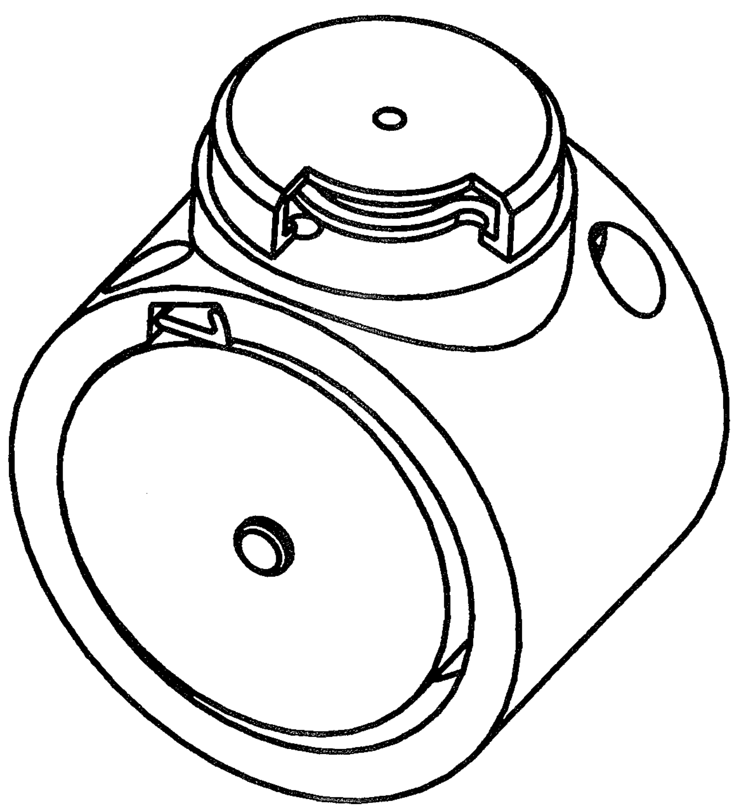


Figure 20. Rotating combustion chamber engine

is transferred from the compression stroke to the combustion chamber and subsequently to the expansion chamber through ports that are timed by the rotating combustion chamber, which also serves as a rotary valve.

The study done by SwRI utilized thermodynamic cycle simulation to examine the invention and determine its potential as an alternative power plant. The study found that its significant advantages were as follows:

- Separate compression and expansion rotors allow the expansion ratio to be substantially higher than the compression ratio, permitting more complete utilization of the heat energy.

- Port timings can be arbitrarily established to optimize the gas exchange processes.

- Constant or near-constant volume combustion can be achieved by port timing.

- The duration at constant volume can be tailored to the combustion process.

- Perfect balance assures that the engine can operate at high speed.

- The engine achieves a power stroke on every revolution while retaining the benefits of a four-stroke cycle.

While these benefits are important, there are also some significant technical risks:

- The sealing challenge is more significant than for the Wankel engine because of a large number of sealing surfaces.

- The seal between the rotor and housing depends upon close tolerances and cannot permit a floating seal member because of the need to pass under a floating vane.

- The separate expansion rotor will be highly thermally loaded.

- The dynamics of the floating vanes may limit speed.

- The gas exchange process also limits the speed at which good efficiency can be maintained.

The SwRI study only investigated the thermodynamic cycle and did not address sealing, lubrication, or cooling issues, which are the significant technical challenges. While the study predicted power density to be substantially higher than for conventional automotive engines, the thermal efficiency was predicted to be lower, owing primarily to large amounts of heat transfer due to large surface area. This concept is attractive because of its potential for high power density at high shaft speed; however, it is also extremely high risk at its current conceptual stage. Working prototypes have not been built.

Based on the estimates of SwRI's study, the engine would have specific volume of about 1.5 L/kW and specific weight of about 1.2 kg/kW. At a rated speed of 10,000 rpm, the corresponding generator would have specific volume at roughly 0.22 L/kW and weight of about 0.8 kg/kW. Thermal efficiency of around 19 percent was predicted for the engine, for an overall system efficiency of 17.5 percent. Cost of the engine may be fairly high because of high-temperature materials requirements. Emissions should be equivalent to the four-stroke engines. Producibility and reliability would be questionable due to expensive materials and hot environment of the combustion chamber and expansion rotor. This concept has very high technical risk. NVH should be well controlled due to good balance. Multifuel capability is equivalent to SI engines, and transient response should be fairly good due to low inertia. Starting may be difficult because of the high leakages around the large sealing surfaces and resulting poor compression at low speed.

3. HCCI Engine With Pressure Relief

Homogeneous Charge Compression Ignition (HCCI) is an unconventional combustion process wherein a premixed charge of fuel and air is ignited by compressing it to the point of autoignition. It is similar in many respects to detonation or knock, the uncontrolled combustion that can damage or destroy SI engines. However, it differs in that an engine designed for HCCI does not necessarily initiate combustion by a spark and does not experience the travel of a flame through the mixture, compressing the unburned charge to the point of detonation. Nor is HCCI generally initiated at hot surfaces of the combustion chamber, as is knock. Rather, the ignition is initiated at a multitude of points within the combustion chamber volume by the presence of active radicals, or combustion precursors. Combustion progresses rapidly and consumes the entire mixture within a short period of time. The intense combustion process has discouraged most researchers from pursuing HCCI as a feasible commercial process; however, if a means of dealing with the extreme rates of pressure rise can be devised, there are many benefits to be gained from HCCI.

Evan Guy Enterprises, Inc., of San Antonio, Texas, has demonstrated a small two-cylinder engine with several unique features devised to take advantage of HCCI for the combustion of heavy distillate fuel. The engine is targeted at the UAV application and operates on a two-stroke cycle. The key feature of the engine is a proprietary cylinder head device designed to soften the combustion process by limiting the peak pressure. Energy is stored mechanically during the combustion process and returned to the cylinder during the expansion stroke for useful work. The engine has been successfully demonstrated burning several fuels, including diesel, JP-8, and gasoline. Its efficiency is relatively high as a result of high effective compression ratio and an effective combustion process, which works well with lean mixtures. Durability problems encountered in early designs have largely been solved.

Based on the limited test data obtained to date, the developed engine has predicted fuel consumption of 0.3 kg/kW·h, for a brake thermal efficiency of 0.275. Its operating speed is relatively high at 6,000 rpm, which will lead to compact generator configuration. The expected specific weight and volume of a developed engine are 1.1 kg/kW and 2.1 L/kW, respectively.

The cost should be advantaged relative to conventional four-stroke engines. In its current configuration, the engine admits a premixed fuel-air charge, so it is subject to emissions problems due to scavenging losses. However, direct injection is an ultimate goal, so emissions should be better than for direct-injected SI two-strokes owing to the clean combustion process. A particular emissions benefit results if the HCCI process can be used to take advantage of the low NO_x possible with extremely lean mixtures, a strong possibility with this engine. Producibility should be good, but obtaining automotive class reliability will require significant development. Noise characteristics should be comparable to competing two-stroke engines. The multifuel capability is very good with volatile fuels but dependent upon preheating of the fuel when running with diesel or other heavy fuels. Transient response is good; however, because of the required fuel preheating, startability is not as good. This should be considered a high risk option since the technology is in early development stages.

4. High-Speed Detonation Engine

It has been observed in the racing industry that engines running at very high crankshaft speed (13,000 to 15,000 rpm) are relatively insensitive to fuel octane rating. Despite high compression ratios, they do not suffer from knock problems, and they are not adversely affected by fuels of octane ratings that would be intolerable in engines of similar compression ratio at conventional speeds. The explanation is that at these high speeds, the expansion process is so fast that the gas is expanded before significant pressure excursions can occur. The combustion process is largely a detonation process, similar to HCCI, but no component damage is encountered because of the rapid expansion. Since high speed is also desired for the generator of an APU, this approach to the engine and combustion system may be of interest for the APU system.

The chief advantage of the high-speed detonation engine (HSDE) over conventional technology is high power density owing to high crankshaft speed. The disadvantages are several, including durability, thermal efficiency due to high friction, noise, and risk. Scores for the high-speed detonation engine are based on a three-cylinder engine of 60-mm bore and 50-mm stroke, for a total displacement of 0.43 L, running at 13,000 rpm. Cycle simulation predictions with this engine indicate that it could produce 30 kW·h assuming a generator efficiency of 92 percent.

Predicted thermal efficiency of the engine was 24.6 percent, lower than conventional four-stroke engines primarily because of friction and pumping losses at high speed. With generator, a thermal efficiency of 22.6 percent is predicted. The engine could be packaged with a specific volume of about 1.75 L/kW and specific weight of 0.7 kg/kW, assuming weight scales with volume from conventional engines.

Cost of the HSDE should be comparable or slightly higher than a conventional four-stroke engine. Emissions should also be comparable or slightly worse, as alteration of valve events to achieve good volumetric efficiency at high speed may cause increased hydrocarbon emissions. Producibility will be similar, but reliability will be markedly worse. This is a high risk option, since the technologies are currently only used in racing engines that do not have to meet cost and durability constraints. Noise would be worse than with conventional technology. The multifuel capability would be slightly better than with conventional SI engines, since the octane requirements would be reduced. Transient response should be fairly good, although it may take the engine longer to reach the high speeds at which it is designed to operate. Startability will be limited by high compression ratios, requiring relatively high cranking torque.

5. Model Airplane Engine

Another type of engine which utilizes a homogeneous compression-ignition type combustion system are the small engines used for model airplanes. These engines typically run at very high rpm enabled by small displacement, short stroke, and low reciprocating mass, and use either alcohol fuel or diesel fuel heavily doped with additives to control knock and assure stable combustion. They are attractive from the standpoint of high power-to-weight ratio and high shaft speed, but currently only exist in small power ratings, generally of 10 kW and below. Two possibilities exist to use this technology for hybrid APUs: 1) develop a multicylinder engine with enough cylinders to make 30 kW (roughly 12 to 15 cylinders would be needed), or 2) develop a smaller APU and equip the vehicle with several model airplane engines to achieve the desired power rating. The second option is intriguing because the individual package volume would be small and several units could conceivably be distributed to various locations in the vehicle that would otherwise be unutilized. It is also of interest because of a general need for small,

lightweight motor-generator units for applications other than hybrid vehicles, particularly for military use. Hence, the scoring of this concept for the hybrid vehicle will be based on the second option.

The key to success of these engines for hybrid APUs will be the development of a combustion system that burns conventional fuels and that is nonpolluting. The model airplane engine is typically a carbureted two-stroke, and thus unacceptable for vehicle emissions. Direct fuel injection is difficult to accomplish in the scale required for the small cylinders; therefore, the most likely option is a four-stroke engine. There are many model airplane four-stroke engines currently in production, so the technology already exists, but the power density will suffer somewhat in comparison with the two-strokes. To achieve conventional fuel compatibility, a spark-ignited combustion system is the likely candidate, with gasoline or natural gas fuel. HCCI might also be contemplated, at significant added risk.

Estimates for scoring are developed primarily from advertising information of model airplane engine manufacturers. An engine power-to-weight ratio of 0.6 is probably achievable, along with specific volume of 1 L/kW. Fuel consumption is likely to be significantly worse than for automotive technology engines, as these engines typically use a rich fuel-air mixture; however, an engine designed for the hybrid APU application could improve greatly upon the current fuel consumption of this class of engine. BSFC is estimated at 0.35 kg/kW·h, resulting in thermal efficiency of 24 percent. Assuming the engine turns at 10,000 rpm, the generator specific volume and weight are estimated to be 0.22 L/kW and 0.86 kg/kW, respectively. Cost should be low in mass production. Emissions are questionable and would require significant development work. Producibility is good, but reliability would also require significant development, as model aircraft engines are not currently designed for long life. This is a high risk option due to the immaturity of the technology for this application. Noise is a significant issue for high rpm engines but can be dealt with primarily by mufflers. Multifuel capability would be equivalent to conventional SI engines. Transient response would be better because of low inertia, and cranking torque should be low.

6. Two-Stroke Gas Generator With Turboalternator

As an option to the FPE gas generator driving a turboalternator, a fairly conventional two-stroke engine could also serve as the gas generator. In this system configuration, the two-stroke engine would provide shaft power to a compressor, which may be a rotating or reciprocating machine. The compressor would boost the scavenging air for the two-stroke cycle to an elevated pressure, where it would then pass through the engine. The exhaust of the engine would be routed to the turboalternator. This concept is shown schematically in Fig. 21.

A cursory analysis of this concept was done using a cycle simulation program for externally scavenged two-stroke engines, backpressuring the engine with an orifice restriction to represent the turbine. Turbine power was calculated based on the developed pressure and temperature in the exhaust manifold, assuming 80 percent efficiency. The engine was driving a compressor at 77 percent efficiency. This analysis determined that the appropriate flowrates for a 30-kW

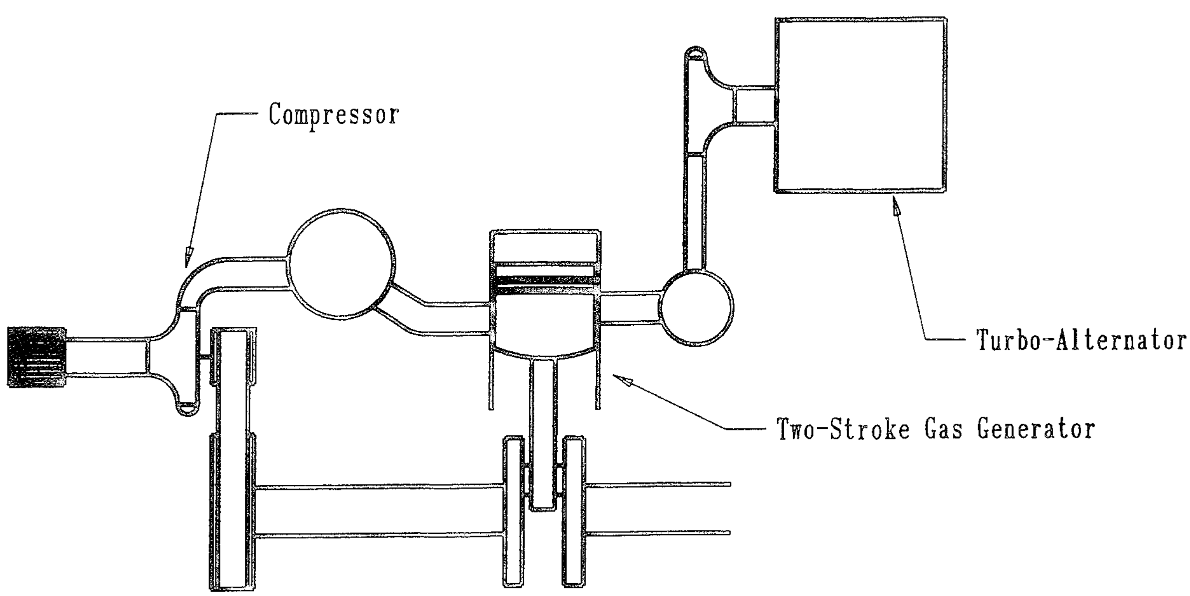


Figure 21. Conceptual design of two-stroke gas generator/turboalternator

generator could be achieved with a three-cylinder engine of about 0.8-L total displacement. Thermal efficiency of the complete cycle was predicted to be 22 percent. This engine would have a specific weight of around 1.2 kg/kW and specific volume of about 2.6 L/kW. Its scores for specific weight and volume are not as good as for the conventional SI two-stroke engine because to achieve the proper balance of airflow and pressure rise for the turboalternator, a lean-burn engine condition was assumed. Also, the engine deals with a proportionally larger volumetric flow of air than an equivalent sized conventional engine, so there is a penalty for the air handling ducts and filters. It would benefit from the generator size and weight advantages of high speed.

Cost for this system would be about the equivalent of a conventional four-stroke engine/generator; cost advantages of the simple two-stroke engine would be offset by the added cost of high speed turbomachinery. Emissions should be good, as a lean-burn combustion system will lead to low NO_x emissions. Producibility should be comparable with conventional two-stroke engines, and reliability should be good, as the turbine inlet temperature is very low. There is some technical risk associated with the development of an engine tailored to this application, but the technologies are all fairly well developed. Noise should be better than with conventional engines because of the noise suppression effects of the turbine. Multifuel capability is equivalent to conventional engines. Transient response is somewhat worse than conventional engines because of the turbine inertia. In starting, the ease of cranking the two-stroke engine may be offset by the reliance upon the compressor for scavenging air.

7. Regenerative Internal Combustion Engine

SwRI maintains an interest in more advanced engine concepts, and has performed studies on a number of concepts related to Stirling cycle engines. The promise of these ideas is increased thermal efficiency and reduced emissions; however, they have always been associated with high weight, volume, and cost penalties, as well as technical risk, and have not yet become viable commercial products. One such concept was investigated in an internal research project in 1987 and has since been refined.(42) It has now been named the Regenerative Internal Combustion Engine (RICE). A thermodynamic analysis of the engine cycle predicted an indicated thermal

efficiency of 44 percent. Indicated power of 84 kW was predicted from a single-working cylinder engine of 0.8-L displacement. The analysis was based on no-loss flow conditions and idealized timing of valve events; more realistic estimates would reduce the efficiency and power output somewhat.

The RICE concept is illustrated in Fig. 22. Two cylinders are employed, one serving as the compressor and one as the expander. A transfer valve controls flow out of the compressor and into a regenerator volume, into which a matrix of heat absorbing material is placed. The purpose of the regenerator is to recover thermal energy from the burned products after the expansion event, transferring this energy to the compressed gas from the compressor prior to combustion. Thus, a portion of the heat energy supplied to the gas before expansion comes from the previous burned charge, reducing the amount of thermal input needed from combustion. Upon completion of the expansion event, the burned charge is pumped back through the regenerator and out the exhaust valve.

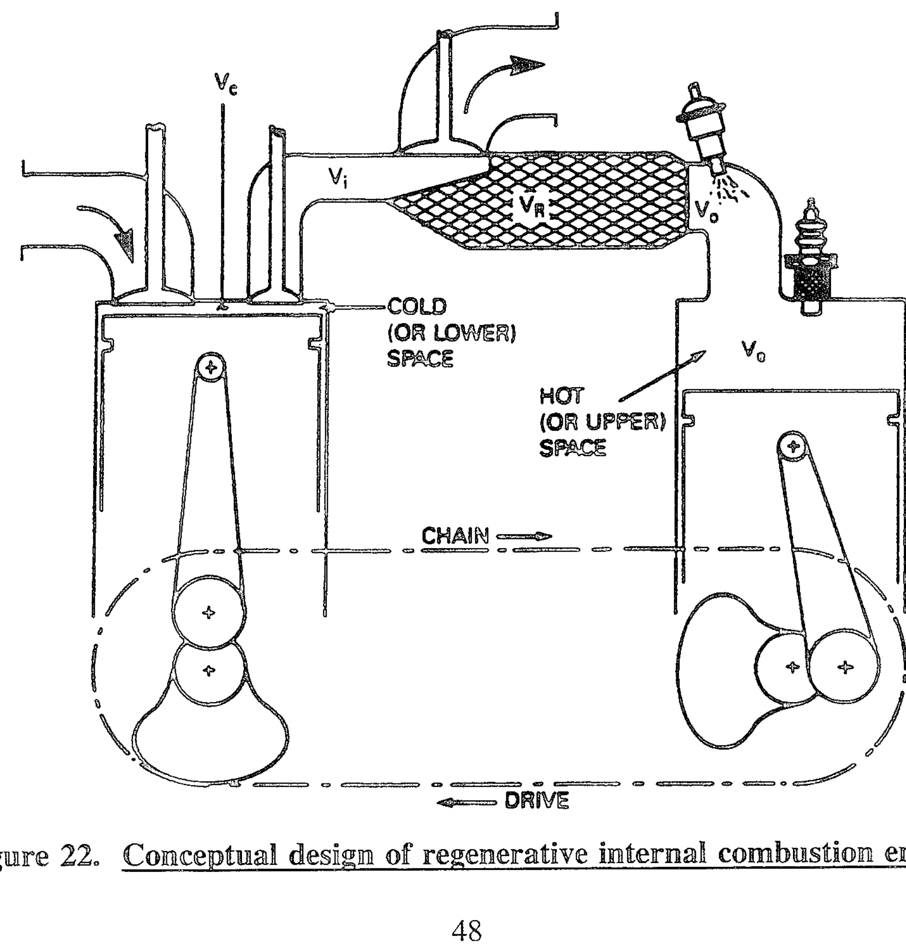


Figure 22. Conceptual design of regenerative internal combustion engine

Accounting for realism in fluid flow losses, valve events, and dead volumes, as well as mechanical friction, the brake thermal efficiency of this device may be as high as 33 percent. An APU utilizing this concept would then have brake thermal efficiency of 30.4 percent. It is difficult to predict weight and volume from a conceptual design, but it could be expected to be heavier and bulkier than conventional engines for a given working displacement. The first analysis, however, predicted a very high power density based on displacement. Accounting for the realistic inefficiencies, the displacement of the working cylinder needed for a 30-kW power plant is about 0.41 L, and the engine could be well represented by a two-cylinder engine of 0.82 L total displacement. This engine may be expected to have a weight of 50 kg and volume of 82 L, giving specific weight and volume of 1.7 kg/kW and 2.7 L/kW, respectively. The generator weight and volume would be the equivalent of conventional systems.

Cost will be relatively high, owing primarily to the regenerator, which will require high temperature materials. Some development of the fuel injection and combustion system will be needed to assure low emissions, and NO_x emissions may be a problem due to preheating of the air or air/fuel mixture. Producibility and reliability may be slightly worse than for conventional engines, owing again to the regenerator. This is a relatively high technical risk approach. NVH is probably better than for conventional engines, as the regenerator will serve to reduce exhaust pulsation. Multifuel capability is unclear, as the combustion system is as yet unspecified, but is probably better than conventional spark-ignited engines since the charge is preheated. The transient response characteristics will be worse than for conventional engines, and starting will be slow with the need to warm up the regenerator.

B. Subsystem Concepts

For the purpose of ranking the subsystem concepts, they are considered as applied to the baseline engine, the four-stroke spark-ignited engine. Thus, they can be compared on an equivalent basis. The scores are expressed as increments to the baseline. Most of the ideas are applicable to several of the system concepts.

1. Electric Valves

A limitation to the speed and performance of conventional poppet-valved four-stroke engines is the valvetrain, both in terms of valve dynamics limiting engine speed and fluid dynamics producing pumping losses for flow into and out of the cylinders. Aura Systems, Inc. has patented an electromagnetic valve actuation system which helps to overcome some of these limitations.(41) Its claimed advantages are

- elimination of the conventional valvetrain for reduced engine friction;
- low electric power consumption (53 W per valve on a 16-valve engine at 7,500 rpm);
- rapid valve actuation, achieving near square-wave valve motion; and
- variable, programmable valve timing.

With further development, the system could likely increase the speed capability of conventional engines. Aura Systems predicts increased power by 10 to 20 percent, increased fuel economy by 10 to 20 percent, and emissions improvement in comparison with conventional SI engines. These predictions seem plausible if the system does mechanically what they claim. Even more increase in power density could be achieved if the engine rated speed is increased. The developed system cost in production is probably equivalent to a conventional valvetrain; the added electromagnetic components and power electronics replace the conventional camshaft and valvetrain components. NVH may be reduced by the elimination of valvetrain mechanical noise. Transient response may also be improved by programmable valve timing. An automatic compression release function could be programmed into the valves to aid in starting. There are technical risks associated with this technology.

2. Stepwise Mixture Control and Turndown

Conventional emissions control in a SI engine depends upon a three-way catalyst and tight control of fuel-air ratio around the stoichiometric condition. If the fuel-air ratio deviates far from stoichiometric on either side, HC and CO or NO_x emissions are adversely affected by poor catalyst performance. However, if the engine is operated far enough to the lean side, NO_x emissions are again reduced by virtue of lower combustion temperatures. One approach to emissions control is to operate the engine at stoichiometric fuel-air ratio for rated condition and reduce power by a combination of throttling and lean mixture. If stepwise power transients are desired, as is likely the case with a hybrid APU, this control mode is feasible; however, it is probably not useful for a direct-drive vehicle. The advantages of this concept are improved part-load fuel economy and emissions, which may or may not be a strong driver for the hybrid APU. There are no other perceived advantages.

3. Step-Up Gearbox

To take advantage of the reduced weight and size associated with generators running at high shaft speed, one approach is simply to employ a step-up gearbox. The tradeoffs of this approach are increased size, weight, cost, and reduced efficiency associated with the gearbox, with reduced size and weight of the generator. This idea is applicable to any crankshaft engine. Estimates for the relative effects of incorporating this idea are based on the assumption of a 4:1 step-up gear ratio to achieve 24,000 rpm with the engine speed at 6,000 rpm. The gearbox for this application would likely be a two-step geartrain with 2 to 1 ratios on both gear sets. Its weight and volume would be about 20 kg and 12 L, respectively. The torque losses would be about 3 percent, resulting in an effective generator efficiency of 89 percent. Slight penalties would be incurred in system cost, noise, startability, and transient response.

4. Step-Up Gearbox Integrated With Crank

To limit the penalties associated with a step-up gearbox, an engine could be designed to incorporate the features of the gearbox into the crankshaft system, thus reducing somewhat the weight, volume, and cost penalties. Other effects on system characteristics would be unchanged.

5. Blowdown Capture Turbocharging

Proponents of turbocharging have long recognized the importance of exhaust system pressure pulsations in the performance of the turbine. Pulse effects are often credited for improvements in turbine efficiency on the order of 10 to 30 percent, and "effective" efficiency values based on mean exhaust manifold conditions are often greater than one. Recognizing that a typical reciprocating engine wastes a significant proportion of the exhaust gas energy in the process of blowing down the cylinder from its final expansion pressure to the manifold pressure, SwRI engineers have theorized a means of capturing this otherwise wasted energy. The concept is to utilize separate exhaust valves for the blowdown process and for the exhaust stroke, capturing the higher pressure exhaust products in a separate manifold and achieving higher availability of exhaust energy for work in the turbine. An additional feature of the concept is the use of acoustic elements in the blowdown manifold to further enhance the recovery of blowdown energy. While this concept primarily benefits large, highly turbocharged engines, it also has potential applications to the APU.

One possibility for application to hybrid APUs is to simply highly turbocharge a conventional engine. This could result in improvements in power densities on the order of 30 to 50 percent over the naturally aspirated engine; however, turbocharging is difficult for smaller engines. Fuel economy would also be improved. The improvement attributable to blowdown capture is unclear without further study but could be on the order of 10 percent in power density and 5 percent in fuel economy. Another option would be to take an otherwise naturally aspirated engine and capture the blowdown pulses in a separate manifold, routing them to a turboalternator. This would be of special interest in parallel drivetrains where the shaft power of the engine would be used for direct drive and the turbine power for charging the batteries. Further investigation is

perhaps merited for this option but outside the scope of consideration in this study. Risks associated with the idea are primarily to the durability of the turbine, which will be exposed to higher temperature air.

For the purposes of ranking in this study, the application to the conventional turbocharged engine is assumed, with the benefits noted above. The applicability of the concept depends upon whether a turbocharged engine is indicated, which is highly dependent upon the required power rating. For smaller vehicles, turbocharging is probably not a reasonable option because of the unavailability of commercial turbochargers in an appropriate size. However, for power ratings of 50 kW or above, turbocharging is a possibility.

6. Combined Cycle Heat Recovery

The primary source of wasted energy in an internal combustion engine is thermal energy in the exhaust system. For SI engines, this energy is of relatively high quality (high temperature) and could be utilized in a combined cycle, similar to cogeneration. This option, referred to as a "bottoming cycle," has been studied by several researchers.(43-46) The technologies to do this include Rankine cycle machines, Brayton cycle machines and Stirling engines, as well as other possibilities. The technologies are not well developed for automotive prime movers and are likely to have significant cost, weight, and volume penalties. Estimates made by one study indicated a potential improvement of about 15 percent in fuel economy at rated conditions for a baseline diesel engine.(43) Volume and weight penalties were not given, but it is assumed for this study that the additional equipment would require about 50 percent of the baseline engine volume, weight, and cost.

VII. DISCUSSION OF CONCEPT RANKING

The comparison analysis of APU system concepts is summarized in TABLE 1. This table presents the raw scores on each of eleven criteria, the calculation of scoring statistics to establish

TABLE 1. Ranking of Hybrid APU System Concepts

RAW SCORES BY CRITERION*:	1	2	3	4	5	6	7	8	9	10	11	12	Composite Score
Free-Piston with Linear Generator	0.30	0.56	0.40	1.40	0.80	1.20	1.40	2.00	0.50	1.20	1.20	2.00	
Wankel Rotary Engine	0.25	1.32	0.53	1.30	0.90	1.00	1.10	1.50	0.90	1.10	1.10	1.10	
Two-Stroke SI Engine	0.27	0.42	0.48	1.30	0.80	1.20	1.20	1.20	0.90	1.00	1.00	1.10	
Two-Stroke Gas Generator/Turboalternator	0.22	0.38	0.70	1.00	1.10	1.00	1.20	1.00	0.80	1.10	1.00	0.80	
Free-Piston with Turboalternator	0.17	0.49	0.50	1.20	0.80	1.00	1.10	2.00	0.90	1.30	1.20	0.90	
HCCI Two-Stroke with Pressure Relief	0.25	0.42	0.38	1.20	1.20	1.00	0.70	0.80	0.50	1.00	1.10	1.10	
Two-Stroke CI Engine	0.30	0.41	0.36	1.20	0.70	1.10	1.10	0.70	0.80	0.90	0.80	1.10	
Multicylinder Model Airplane Engine	0.22	0.82	0.68	1.00	0.70	0.60	0.60	1.05	0.60	0.80	1.00	1.20	
Four-Stroke SI Engine	0.27	0.30	0.37	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	
High Speed Detonation Engine (HSDE)	0.23	0.52	0.68	0.95	0.95	0.50	0.50	0.80	0.50	0.50	1.10	0.95	
Four-Stroke CI Engine	0.33	0.23	0.21	1.00	0.90	1.10	1.10	0.50	1.00	0.80	0.80	0.80	
Regenerative Internal Combustion Engine	030	0.33	0.31	0.85	0.90	0.95	0.95	0.50	0.60	1.10	1.10	0.90	
Rotating Combustion Chamber Engine	0.18	0.58	0.50	0.90	1.00	0.50	0.50	0.70	0.30	1.10	1.00	1.10	
SCORING STATISTICS													
Mean	0.25	0.52	0.47	1.10	0.90	1.02	0.96	1.06	0.72	0.99	1.03	1.08	
Max	0.33	1.32	0.70	1.40	1.20	1.20	1.40	2.00	1.00	1.30	1.20	2.00	
Min	0.17	0.23	0.21	0.85	0.70	0.80	0.50	0.50	0.30	0.50	0.80	0.80	
Significance	4	5	5	4	4	3	3	2	1	2	1	2	
Normalizing Factor	6.20	0.92	2.02	1.82	2.00	2.50	1.11	0.67	1.43	1.25	2.50	0.83	
Weight	24.81	4.60	10.08	7.27	8.00	7.50	3.33	1.33	1.43	2.50	2.50	1.67	
WEIGHTED SCORES BY CRITERION:													Composite Score
Free-Piston with Linear Generator	7.44	2.56	4.03	10.18	6.40	9.00	4.67	2.67	0.71	3.00	3.00	3.33	57.00
Wankel Rotary Engine	6.20	6.06	5.36	9.45	7.20	7.50	3.67	2.00	1.29	2.75	2.75	1.83	56.06
Two-Stroke SI Engine	6.62	1.95	4.85	9.45	6.40	9.00	4.00	1.60	1.29	2.50	2.50	1.83	51.99
Two-Stroke Gas Generator/Turboalternator	5.46	1.74	7.10	7.27	8.80	7.50	4.00	1.33	1.14	2.75	2.50	1.33	50.93
Free-Piston with Turboalternator	4.22	2.25	5.04	8.73	6.40	7.50	3.67	2.67	1.29	3.25	3.00	1.50	49.51
HCCI Two-Stroke with Pressure Relief	6.28	1.92	3.88	8.73	9.60	7.50	2.33	1.07	0.71	2.50	2.75	1.83	49.10
Two-Stroke CI Engine	7.53	1.87	3.60	8.73	5.60	8.25	3.67	0.93	1.14	2.25	2.00	1.83	47.41
Multicylinder Model Airplane Engine	5.48	3.77	6.91	7.27	5.60	7.50	2.00	1.40	0.86	2.00	2.50	2.00	47.29
Four-Stroke SI Engine	6.62	1.39	3.73	7.27	8.00	7.50	3.33	1.33	1.43	2.50	2.50	1.67	47.28
High Speed Detonation Engine (HSDE)	5.61	2.38	6.91	6.91	7.60	7.50	1.67	1.07	0.71	1.25	2.75	1.58	45.93
Four-Stroke CI Engine	8.22	1.06	2.10	7.27	7.20	7.50	3.67	0.67	1.43	2.00	2.00	1.33	44.44
Regenerative Internal Combustion Engine	7.54	1.53	3.15	6.18	7.20	7.13	3.17	0.67	0.86	2.75	2.75	1.50	44.43
Rotating Combustion Chamber Engine	4.34	2.68	5.04	6.55	8.00	6.00	1.67	0.93	0.43	2.75	2.50	1.83	42.72
MEAN WEIGHTED SCORE:	6.27	2.40	4.75	8.00	7.23	7.64	3.19	1.41	1.02	2.48	2.58	1.80	48.7

*CRITERIA:

1.	Thermal Efficiency	2.	Volume (Power density)	3.	Weight (Power density)
4.	Cost (as product)	5.	Emissions	6.	Producibility
7.	Reliability	8.	Cranking Torque and Startability	9.	Technical Risk
10.	Noise, Vibration, Harshness	11.	Multifuel Capability	12.	Transient Response

final weighting, and the weighted scores. Composite scores at the right of the table are the sum of weighted scores for all criteria. The concepts are ranked in order of their score, from highest to lowest. The mean weighted score is provided as a means of evaluating the general merit of a concept.

The highest scoring concept was the free-piston engine linear generator (FPELG). The primary contributors to this ranking were perceived advantages in thermal efficiency, cost, producibility, reliability, NVH, fuel tolerance, and transient response. It is less attractive from the standpoints of power density and risk; however, it still ranks ahead of the baseline four-stroke SI engine in the power density categories. It should be noted that the power density of the FPELG is still unknown and could be grossly overestimated, hence the technical risk. This concept certainly merits detailed investigation.

The next highest ranked concept is the Wankel rotary engine. By contrast with the FPELG, its advantages are primarily in the area of power density and risk, with lesser (but still substantial) benefits in terms of cost and NVH. The Wankel engine is well-established and offers a low risk alternative for advanced high power-density APUs.

These top two concepts were very closely ranked. Considering the subjectiveness of this analysis, their order of ranking should be considered interchangeable. By contrast, the next highest ranked concept, the two-stroke SI engine, scored significantly lower. It scored well in the categories of thermal efficiency, cost, producibility, reliability, and risk. Interestingly, it did not score as well in power density. Although the two-stroke engine has been touted as a high power-density alternative, its advantages are weakened by the need to consider the weight of a generator in the system. The two-stroke engine has some advantages in the respect that it can run at higher rpm than a conventional engine; however, the Wankel engine is even better at achieving high power in a small package.

The combination of a two-stroke gas generator with a turboalternator scored fairly well, primarily owing to a high power-to-weight ratio and a fairly well developed state of technology for the

system components. It is disadvantaged with respect to thermal efficiency, cost, and volumetric power density, the latter attributable to the large air handling requirement.

The free-piston engine/turboalternator combination scored about 1.4 points below the conventional two-stroke engine gas generator with turboalternator. This lower score is attributed to lower thermal efficiency, weight power density, and emissions. The emissions difference is debatable and is based on the presumption of a lean-burn cycle for the conventional crank-driven two-stroke. The lean-burn conditions could also be applied to the free-piston engine, making its score equivalent to the crank engine. Both options should be investigated further to better quantify the differences, particularly in power density.

The HCCI two-stroke engine with pressure relief scored ahead of the mean overall but is seen as a high risk approach. It has the potential of low cost, good thermal efficiency, and inherently low NO_x emissions. Its score is very close to that of the FPE-turboalternator, suggesting that further investigation is also warranted.

The composite scores of remaining concepts were below the average. Many of these ideas are good in one or two respects but suffer in one or several key areas needed to achieve the overall goals of a hybrid APU. The primary discriminators are thermal efficiency and power density.

The scoring summary for the auxiliary concepts is shown in TABLE 2. For these ideas, the composite score represents the overall ability to improve over the state-of-the-art for the hybrid application. Only two of the six concepts scored positive: electric valves and stepwise mixture control. The potential benefits of electric valves are strong; their main drawback is technical risk. The stepwise mixture control approach can be implemented by a control system strategy, without additional cost. Therefore, it offers a low risk technique for improved emissions and thermal efficiency.

TABLE 2. Ranking of Hybrid APU Auxiliary Concepts

RAW SCORES BY CRITERION*:	1	2	3	4	5	6	7	8	9	10	11	12	
Electric Valves	0.04	0.05	0.03	0.00	0.10	0.00	0.00	0.20	-0.20	0.10	0.00	0.05	
Stoich. NG 4-Stroke w/Lean Turndown	0.02	0.00	0.00	0.00	0.10	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
Step-up Gearbox Integrated w/Crank	-0.01	-0.01	0.08	-0.03	0.00	0.00	0.00	-0.05	0.00	-0.05	0.00	-0.05	
Step-Up Gearbox	-0.01	-0.02	0.06	-0.05	0.00	0.00	0.00	-0.05	0.00	-0.05	0.00	-0.05	
Blowdown Capture Turbocharging	0.01	0.03	0.02	-0.10	0.00	-0.10	-0.10	0.00	-0.10	0.10	0.00	-0.10	
Combined-Cycle Heat Recovery	0.04	-0.09	-0.07	-0.20	0.00	-0.10	-0.20	0.00	-0.30	0.10	0.00	-0.10	
SCORING STATISTICS													
Mean	0.02	-0.01	0.02	-0.06	0.03	-0.03	-0.05	0.02	-0.10	0.03	0.00	-0.04	
Max	0.04	0.05	0.08	0.00	0.10	0.00	0.00	0.20	0.00	0.10	0.00	0.05	
Min	-0.01	-0.09	-0.07	-0.20	0.00	-0.10	-0.20	-0.05	-0.30	-0.05	0.00	-0.10	
Significance	4	5	5	4	4	3	3	2	1	2	1	2	
Normalizing Factor	19.99	7.02	6.79	5.00	10.00	10.00	5.00	4.00	3.33	6.67	n/a	6.67	
Weight	79.97	35.08	33.94	20.00	40.00	30.00	15.00	8.00	3.33	13.33	n/a	13.33	
WEIGHTED SCORES BY CRITERION:													Composite Score
Electric Valves	3.20	1.68	0.90	0.00	4.00	0.00	0.00	1.60	-0.67	1.33	0.00	0.67	12.71
Stoich. NG 4-Stroke w/Lean Turndown	1.60	0.00	0.00	0.00	4.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	5.60
Step-up Gearbox Integrated w/Crank	-0.80	-0.35	2.71	-0.60	0.00	0.00	0.00	-0.40	0.00	-0.67	0.00	-0.67	-0.77
Step-Up Gearbox	-0.64	-0.58	1.87	-1.00	0.00	0.00	0.00	-0.40	0.00	-0.67	0.00	-0.67	-2.08
Blowdown Capture Turbocharging	1.07	1.06	0.58	-2.00	0.00	-3.00	-1.50	0.00	-0.33	1.33	0.00	-1.33	-4.12
Combined-Cycle Heat Recovery	2.94	-3.32	-2.29	-4.00	0.00	-3.00	-3.00	0.00	-1.00	1.33	0.00	-1.33	-13.66
MEAN WEIGHTED SCORE:	1.23	-0.25	0.63	-1.27	1.33	-1.00	-0.75	0.13	-0.33	0.44	0.00	-0.56	-0.39

*CRITERIA:

1.	Thermal Efficiency	2.	Volume (Power density)	3.	Weight (Power density)
4.	Cost (as product)	5.	Emissions	6.	Producibility
7.	Reliability	8.	Cranking Torque and Startability	9.	Technical Risk
10.	Noise, Vibration, Harshness	11.	Multifuel Capability	12.	Transient Response

VIII. RECOMMENDATIONS

All of these concept scorings and rankings are subject to substantial error. Many of the concepts have only been subjected to cursory evaluation and limited analysis, whereas others represent near production-ready technology. This is reflected in the risk scores. It should be borne in mind that the evaluation is highly subjective in nature and could have a different outcome if a different person was doing the analysis. Nonetheless, it is believed that these results present as fair and unbiased an analysis as can be conducted in a study of this scope.

Bearing in mind the potential for error, it is recommended that these rankings be considered as a screening test. SwRI recommends further study for the top six concepts. Although it appears from this analysis that there are two concepts that clearly rank above the rest, these rankings may change as more detailed information develops. The objective of this study will be to further quantify and confirm their ranking relative to each other. This should be done in two steps as follows:

1. Perform an in-depth cycle simulation study of each concept. The thermodynamic cycle simulation can provide system-level comparisons of performance, fuel economy, and emissions. This will directly refine the scoring in the categories of thermal efficiency and emissions and will provide essential information on key component dimensions to further quantify the power density. Thus, most of the higher significance scoring factors will be refined. At the conclusion of this study, the concepts should be reranked and the rankings studied to adjust downward those concepts which are less competitive. This analysis can be done largely in parallel for all six concepts but may concentrate on the top ranked concepts at first.

2. Perform a preliminary design analysis of each remaining concept to gain further refinement of the volume and weight power density. This design analysis will consist of layouts of all key engine and generator components using a CAD system that will provide component mass properties for weight rollups. It will also include systems analysis to assure that accounting is done for weight and size of necessary accessory

systems. At this point, the scoring estimates for volume and weight can be more accurately determined.

At the conclusion of these analyses, it is likely that one or two concepts will emerge as meriting full-scale development into prototype demonstration systems.

For the subsystem concepts, demonstration tests are the most useful means of further quantifying their benefits. It is recommended that the top two concepts be considered for implementation in demonstration test APU systems. Electric valves can be implemented in a conventional engine to quantify their performance and emissions benefits. The lean turndown approach to achieving fuel economy improvements in a four-stroke NG engine can be implemented by simply modifying a control strategy. It can even be investigated by compiling existing data on engines that can run at both stoichiometric and lean conditions and using these data in an appropriate hybrid vehicle simulation.

This analysis has been very useful in establishing a framework for decision making regarding choices of APU systems for hybrid vehicles. It is necessarily subjective but can be greatly improved by the accumulation of more precise data on the comparison parameters. It is hoped that ARPA finds this approach useful as well.

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APPENDIX

Database of Current Production and Research Engines

Four-Stroke Spark Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
44	Mazda	KL	123.00	2.178		
44	Mazda	KF	104.00	2.826		
44	Mazda	KB	97.00	3.007		
45	Mercedes-Benz	M119 HL	530.00		0.400	0.235
46	Nissan	3-L V6	142.00	2.502		
47	GM	Northstar	216.00	1.227	0.972	
48	Collins	Scotch Yoke	56.67	1.676	1.288	0.277
48	Collins	Scotch Yoke	140.00	0.941	0.821	0.277
49	JUNKERS FLUGZEUG, A.G.	211B	782.98	4.128	0.874	
49	NISSAN MOTOR CO. LTD.	2.960 L VG30DETT	223.71	1.709		
49	NISSAN MOTOR CO. LTD.	4.5 L VH45DE	207.30	2.303		
49	JAGUAR CARS LTD.	5.343 L	202.83	2.631	0.789	
49	LOTUS CARS PLC	910 SE 2.2 L	197.00		0.914	0.300
49	TOYOTA MOTOR CORP.	4.0 L 1UZ-FE	186.42	2.208	1.046	
49	NISSAN MOTOR CO. LTD.	2.988 L VQ	186.00	1.771	0.892	
49	AUDI AG	3.562 L	178.97		1.118	
49	FUJI HVY INDSTR (SUBARU	1.994 L TURBO	161.81		0.908	
49	COSWORTH	MBA	161.00	1.011	0.745	
49	TOYOTA MOTOR CORP.	4.5 L 1FZ-FE	158.00	3.289	1.677	
49	FORD MOTOR CO.	4.605 L	156.60	1.748	1.437	
49	GENERAL MOTORS CORP.	3.786 L	152.87		1.256	
49	VOLVO AB	2.922 L B6304F	150.00		1.167	
49	FORD MOTOR CO.	2.5 L MOD V6	143.17		1.157	
49	FORD MOTOR CO.	4.605 L	141.68	1.932	1.588	
49	NISSAN MOTOR CO. LTD.	VQ 2.495 L	134.00	2.190	1.082	
49	VOLKSWAGEN AG	VR6	129.75	1.218	2.867	
49	VOLVO AB	2.435 L	125.00		1.384	
49	TOYOTA MOTOR CORP.	3.956 L	115.58	4.028	2.249	
49	TOYOTA MOTOR CORP.	3.956 L	114.09	4.080	2.279	
49	FUJI HVY INDSTR (SUBARU	1.994 L	110.33		1.178	
49	VOLVO AB	1.984 L	105.00		1.648	
49	NISSAN MOTOR CO. LTD.	1.998 L SR20DE	104.40	2.461		
49	TOYOTA MOTOR CORP.	3S-GE 1.998 L	101.00		1.347	
49	FUJI HVY INDSTR (SUBARU	2.212 L	96.94		1.248	
49	PEUGEOT,RENAULT,VOLVC	2.664 L	93.21	1.971	1.674	
49	FORD OF EUROPE	ZETA (1.8LHO)	93.00		1.392	
49	TOYOTA MOTOR CORP.	3S-FE 1.998 L	86.00		1.512	
49	TOYOTA MOTOR CORP.	1.998 L	85.76		1.458	
49	GENERAL MOTORS CORP.	2.8 L	85.00		1.805	
49	FUJI HVY INDSTR (SUBARU	1.820 L	80.91		1.458	
49	FORD OF EUROPE	ZETA (1.8LSO)	75.00		1.713	
49	NISSAN MOTOR CO. LTD.	1.497 L GA15	71.30	2.900	1.487	
49	GENERAL MOTORS CORP.	1.991 L	67.10		2.038	
49	NISSAN MOTOR CO. LTD.	1.974 L	65.60	3.124	1.753	
49	GENERAL MOTORS CORP.	1.841 L	65.60		2.079	
49	CHRYSLER MOTOR CORP.	2.213 L	64.50		1.519	
49	FORD MOTOR CO.	2.307 L	63.00		1.810	
49	ZAVODI CREVNA (YUGO)	1.585 L	61.89	0.894	0.638	
49	VOLKSWAGEN AG	026.2	50.00	4.946	2.220	
49	BRIGGS & STRATTON	DM 950(DAIHATSU E	23.12	3.637	2.682	
49	KOHLER ENGINES	CH25 COMMAND	18.40	5.052	2.337	
49	ONAN	P224	17.90		7.095	

Four-Stroke Spark Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
49	BRIGGS & STRATTON	DM 700(DAIHATSU E	17.52	4.797	3.424	
49	ONAN	P220	14.91		3.346	
49	HONDA	GX620	14.91	4.157	2.743	
49	KOHLER ENGINES	CH20 (COMMAND)	14.90	4.307	2.752	
49	KOHLER ENGINES	MV20 (MAGNUM)	14.90		3.960	
49	ONAN	P220V	14.90		7.383	
49	KOHLER ENGINES	M20 (MAGNUM)	14.90		3.960	
49	BRIGGS & STRATTON	350700	13.42		2.481	
49	BRIGGS & STRATTON	350400	13.42		2.481	
49	BRIGGS & STRATTON	422400	13.42		2.951	
49	HONDA	GX610	13.42	4.619	3.048	
49	KOHLER ENGINES	MV18 (MAGNUM)	13.40		4.403	
49	KOHLER ENGINES	M18 (MAGNUM)	13.40		4.403	
49	ONAN	P218	13.40		3.724	
49	KOHLER ENGINES	CH18 (COMMAND)	13.40	4.789	3.060	
49	KAWASAKI	FC540V	12.68		3.431	
49	BRIGGS & STRATTON	402400	11.93		3.319	
49	BRIGGS & STRATTON	303400	11.93		2.716	
49	BRIGGS & STRATTON	326400	11.93		4.049	
49	ONAN	P216V	11.90		4.193	
49	KOHLER ENGINES	M16 (MAGNUM)	11.90		4.916	
49	ONAN	P216	11.90		9.076	
49	KOHLER ENGINES	MV16 (MAGNUM)	11.90		4.958	
49	KOHLER ENGINES	CV14 (COMMAND)	10.50	8.729	3.762	
49	KOHLER ENGINES	CH14 (COMMAND)	10.50	7.527	3.810	
49	KAWASAKI	FC420V	10.44		3.448	
49	KAWASAKI	KF150D	10.44		6.466	
49	ONAN	140	10.40		6.827	
49	KOHLER ENGINES	M14 (MAGNUM)	10.40		5.625	
49	KAWASAKI	FC400V	9.69		3.714	
49	KOHLER ENGINES	CV12.5 (COMMAND)	9.33	9.823	4.234	
49	KAWASAKI	FB460V	9.32		3.862	
49	TECUMSEH	OVXL/C125	9.32		6.223	
49	BRIGGS & STRATTON	260700	9.32		4.104	
49	BRIGGS & STRATTON	290400	9.32		3.476	
49	ONAN	125	9.30		7.634	
49	KOHLER ENGINES	CH12.5 (COMMAND)	9.30	8.499	4.301	
49	KOHLER ENGINES	M12 (MAGNUM)	9.00		6.500	
49	TECUMSEH	OVXL120	8.95		6.480	
49	HONDA	EL5000	8.95		28.721	
49	KOHLER ENGINES	CH11 (COMMAND)	8.20	9.636	4.877	
49	HONDA	EW171	8.20	30.937	29.503	
49	HONDA	WT40X	8.20		18.287	
49	BRIGGS & STRATTON	254400	8.20		3.554	
49	KOHLER ENGINES	CV11 (COMMAND)	8.20	11.177	4.817	
49	KOHLER ENGINES	M10 (MAGNUM)	7.50		7.800	
49	BRIGGS & STRATTON	221400	7.46		3.846	
49	TECUMSEH	TVXL220	7.46		7.641	
49	BRIGGS & STRATTON	243400	7.46		5.836	
49	KAWASAKI	KF100D	7.46		6.169	
49	KAWASAKI	FE290D	6.71		5.006	
49	KAWASAKI	FC290V	6.71		3.651	

Four-Stroke Spark Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
49	BRIGGS & STRATTON	161400	6.71		4.259	
49	BRIGGS & STRATTON	233400	6.71		6.219	
49	KOHLER ENGINES	M8 (MAGNUM)	6.00		5.367	
49	TECUMSEH	TVXL195	5.97		9.548	
49	BRIGGS & STRATTON	195400	5.97		3.533	
49	KAWASAKI	FE250D	5.89		5.092	
49	KAWASAKI	FG300D	5.59		4.559	
49	TECUMSEH	TVXL170	5.22		10.920	
49	TECUMSEH	TVM140	4.47		7.774	
49	KAWASAKI	FE170D	3.95		4.428	
49	KAWASAKI	FA210V	3.88		3.868	
49	KAWASAKI	FA210D	3.88		3.353	
49	TECUMSEH	TVM125	3.73		9.316	
49	BRIGGS & STRATTON	104700	3.73		5.067	
49	KOHLER ENGINES	CH5 (COMMAND)	3.73		5.469	
49	BRIGGS & STRATTON	132200	3.73		3.646	
49	KAWASAKI	FG200D	3.73		6.115	
49	HONDA	EB2200X	3.73		25.480	
49	WISCONSIN ROBIN	W1-185V	3.70		8.649	
49	KAWASAKI	FC150V	3.36		3.874	
49	WISCONSIN ROBIN	W1-145V	3.00		9.333	
49	WISCONSIN ROBIN	WT1-125V	3.00		8.433	
49	KAWASAKI	FG150D	2.68		5.103	
49	HONDA	WD20X	2.61		21.073	
49	KAWASAKI	FA130D	2.31		4.326	
49	BRIGGS & STRATTON	82200	2.24		4.911	
49	US ENGINES INC.	41cc	2.20	3.763	2.546	
49	SHINDAIWA	S45B	1.72	35.893	12.419	
49	SHINDAIWA	SM45P	1.72		10.029	
49	SHINDAIWA	EC350	1.64		10.058	
49	US ENGINES INC.	35cc	1.49		3.688	
49	KAWASAKI	FA76D	1.27		5.759	
49	SHINDAIWA	S25P	0.97		13.204	
49	SHINDAIWA	S20HT	0.67	94.040	17.732	

Four-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
50	Hino	P11C-P-IV	239.0	7.626	3.791	0.185
50	Hino	P11C-P-III	221.0	8.247	4.054	0.184
49	BRIGGS & STRATTON	DM 700(DAIHATSU BUILT)	14.5	5.781	4.608	
49	BRIGGS & STRATTON	DM 950(DAIHATSU BUILT)	19.8	4.254	3.492	
49	CATERPILLAR	3306B DITA	201.3	6.663	4.599	0.208
49	CATERPILLAR	3406	242.4		5.116	
49	CATERPILLAR	3176: 250 hp FAMILY	187.0	5.923	4.717	
49	CATERPILLAR	3176: 325 hp FAMILY	242.0	4.577	3.645	
49	CATERPILLAR	3412	484.7		3.837	
49	CATERPILLAR	3176	242.0	4.577	3.645	
49	CATERPILLAR	3304T0	142.0	6.405	5.289	0.249
49	CATERPILLAR	3406C DITAA	316.9	5.590	4.023	0.202
49	CATERPILLAR	3408 TA	325.0	6.020	4.701	0.220
49	CATERPILLAR	3208 NA	130.5	5.173	10.154	0.219
49	CATERPILLAR	3208 DIT	186.4	4.110	7.671	0.217
49	CATERPILLAR	3408	298.3		4.794	
49	CATERPILLAR	3406B DITA	298.3	5.947	4.375	0.201
49	CATERPILLAR	3412	559.3		3.328	
49	CATERPILLAR	3406	279.6		4.434	
49	CATERPILLAR	3408	354.2		4.037	
49	CATERPILLAR	3176: 275 hp FAMILY	205.0	5.403	4.302	
49	CATERPILLAR	3116	186.0		2.925	
49	CATERPILLAR	3176: 300 hp FAMILY	224.0	4.944	3.938	
49	CUMMINS	NTC400 BCIV	298.3	5.534	8.884	0.201
49	CUMMINS	KTTA19-C	484.7	5.170	7.840	0.206
49	CUMMINS	6B5.9	85.8	5.771	9.970	0.222
49	CUMMINS	6CTA8.3	186.0	4.312	3.258	0.206
49	CUMMINS	N-855-C BCI	175.2	8.254	14.780	0.236
49	CUMMINS	4B3.9	56.7	6.497	11.998	0.227
49	CUMMINS	LTA-10	223.7	5.154	8.627	0.203
49	CUMMINS	N14: ESP1	290.0		4.388	
49	CUMMINS	LT-10-C	167.8	6.350	11.593	0.206
49	CUMMINS	NTC-475	354.0	5.007	7.599	0.207
49	CUMMINS	KTTA38-C	1006.7	5.366	9.189	0.208
49	CUMMINS	VTA28-C	596.6	7.264	9.723	0.215
49	CUMMINS	6CT	160.0	5.250	3.800	
49	CUMMINS	6C8.3	119.0	6.442	4.790	0.218
49	CUMMINS	KTTA-50	1342.2	5.597	8.117	0.214
49	CUMMINS	VT-903-C	279.6	5.357	8.511	0.220
49	CUMMINS	6BTA5.9	134.2	4.020	6.742	0.206
49	CUMMINS	4BTA3.9	89.5	4.633	8.102	0.219
49	DAEWOO HEAVY IND. LTD.	1.8 L(3AB1)	28.0	9.730	7.750	
49	DAEWOO HEAVY IND. LTD.	7.3 L (D0846M)	107.0	7.566	5.794	
49	DAEWOO HEAVY IND. LTD.	2.4 L(C240)	38.0	7.817	5.868	
49	DAEWOO HEAVY IND. LTD.	7.3 L (D0846HM)	124.0	6.528	5.000	
49	DAEWOO HEAVY IND. LTD.	8.1 L (D1146)	133.0	6.226	5.113	
49	DAEWOO HEAVY IND. LTD.	11.1 L (D2366)	165.0	7.485	5.358	
49	DAEWOO HEAVY IND. LTD.	10.4 (D2156HM)	158.0	7.885	5.380	
49	DAEWOO HEAVY IND. LTD.	14.6 L (D2848T)	245.0	5.889	3.633	
49	DAEWOO HEAVY IND. LTD.	19.8 L (MD336)	110.0	32.307	23.091	
49	DAEWOO HEAVY IND. LTD.	21.9 L (D2842T)	364.0	5.202	3.077	
49	DAEWOO HEAVY IND. LTD.	13.2 L (MD334)	74.0	21.658	22.027	

Four-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
49	DAEWOO HEAVY IND. LTD.	21.9 L (D2842L)	429.0	4.414	2.937	
49	DAEWOO HEAVY IND. LTD.	4.8 L(D0844M)	66.0	11.135	7.242	
49	DAEWOO HEAVY IND. LTD.	10.4 L (D2156MT)	188.0	7.156	4.787	
49	DAEWOO HEAVY IND. LTD.	3.3 L(4BC2)	65.0	6.122	4.631	
49	DAEWOO HEAVY IND. LTD.	11.1 L (D2366T)	212.0	5.552	4.358	
49	DAEWOO HEAVY IND. LTD.	19.8 L (MD336TI)	147.0	33.913	17.347	
49	DAEWOO HEAVY IND. LTD.	8.1 L (D1146T)	162.0	5.363	4.444	
49	DAEWOO HEAVY IND. LTD.	2.4 L(C223)	45.0	6.500	4.733	
49	DAEWOO HEAVY IND. LTD.	14.6 L (D2848M)	162.0	6.552	5.185	
49	DAEWOO HEAVY IND. LTD.	5.4 L(6BB1)	99.0	6.515	4.545	
49	DAEWOO HEAVY IND. LTD.	18.3 L (D28480T)	324.0	4.751	3.333	
49	DAF	DKZ-1160	270.7	5.333	7.518	
49	DAF	DKA-1160	173.7	6.418	11.292	
49	DAF	DE385	59.7	8.122	12.019	
49	DAF	DHS 825	190.2	7.090	8.577	
49	DAIMLER-BENZ	OM617	60.4	6.839	9.205	0.269
49	DAIMLER-BENZ	OM364	69.3	6.323	10.656	
49	DAIMLER-BENZ	OM366	105.1		9.330	
49	DAIMLER-BENZ	OM617A	82.0	5.227	6.742	0.249
49	DAIMLER-BENZ	OM366LA	155.1		7.034	
49	DAIMLER-BENZ	OM422	211.0	3.965	8.890	
49	DAIMLER-BENZ	OM616	49.2	5.748	9.103	0.269
49	DALIAN DIESEL ENGINE WOR	CA6110	116.0	8.469	4.828	0.227
49	DETROIT DIESEL	SERIES 60: 11.1L	239.0	7.169	5.088	
49	DETROIT DIESEL	6.2 HD	115.6	3.117	6.065	0.263
49	DETROIT DIESEL	V8-8.2T	171.5	3.532	6.775	0.219
49	DETROIT DIESEL	SERIES 60: 12.7L	298.0	5.741	4.111	
49	FARYMANN DIESEL	18W	5.0	12.906	7.400	0.290
49	FARYMANN DIESEL	75W	17.0	7.922	9.412	0.270
49	FARYMANN DIESEL	36E	8.0	15.619	10.000	0.275
49	FARYMANN DIESEL	29C	8.0	11.587	8.750	0.240
49	FARYMANN DIESEL	41A	9.0	14.532	9.111	0.270
49	FARYMANN DIESEL	18D	5.0	11.062	8.200	0.290
49	FARYMANN DIESEL	32A	8.0	11.587	9.000	0.240
49	FARYMANN DIESEL	32W	9.0	8.814	8.556	0.250
49	FARYMANN DIESEL	86W	22.0	5.877	6.409	0.258
49	FARYMANN DIESEL	71W	16.2	8.407	9.568	0.270
49	FARYMANN DIESEL	95A	18.5	7.747	9.189	0.270
49	FARYMANN DIESEL	57	15.0	7.083	7.333	0.258
49	FARYMANN DIESEL	41E	9.0	13.884	9.444	0.270
49	FARYMANN DIESEL	95W	19.2	7.374	10.156	0.270
49	FARYMANN DIESEL	85	22.0	5.877	6.409	0.258
49	FARYMANN DIESEL	15W	5.0	12.906	7.400	0.300
49	FARYMANN DIESEL	15D	4.0	13.827	9.875	0.300
49	FARYMANN DIESEL	66A	17.0	7.509	7.529	0.239
49	FARYMANN DIESEL	44A	11.0	9.360	9.273	0.239
49	FARYMANN DIESEL	115 W	30.0	5.078	5.433	0.258
49	FARYMANN DIESEL	58W	15.0	7.083	7.333	0.258
49	FARYMANN DIESEL	36A	8.0	14.833	10.250	0.275
49	FARYMANN DIESEL	75A	17.0	7.694	7.941	0.270
49	FARYMANN DIESEL	71A	16.5	7.927	7.879	0.270
49	FARYMANN DIESEL	114	30.8	4.946	5.292	0.258

Four-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
49	FARYMANN DIESEL	21A	6.0	8.908	8.833	0.260
49	FIAT	8340.04	75.3	6.862	12.003	0.214
49	FIAT	8140.61	56.7	3.988	8.487	0.249
49	GARDNER	6LXCT	169.3		11.177	
49	GARDNER	6LXDT	205.2	6.160	9.220	
49	HINO MOTORS	EM100	114.0	8.621	6.491	
49	HINO MOTORS	EK130-T	173.0	9.581	6.069	
49	HINO MOTORS	EP100-T	127.0	8.913	6.083	
49	HINO MOTORS	WO4C-T	103.0	3.648	3.301	
49	HINO MOTORS	WO4D	84.5	4.327	3.716	
49	HINO MOTORS	EF750T	272.0	6.720	4.441	
49	HINO MOTORS	HO6C-T	151.0	5.060	3.616	
49	HINO MOTORS	P11C	239.0	7.626	3.791	
49	HINO MOTORS	EK100	198.0	7.669	4.949	
49	HINO MOTORS	WO6D	110.5	4.430	3.837	
49	HINO MOTORS	EP100-TI	212.5	5.327	7.999	
49	HINO MOTORS	EH700	91.0	8.127	5.495	
49	HINO MOTORS	WO4C-T	84.0	5.171	4.226	
49	HINO MOTORS	EM10U	174.0	8.505	4.828	
49	HINO MOTORS	EH700	121.0	5.903	4.298	
49	HINO MOTORS	P09C	232.0	7.220	3.953	
49	HINO MOTORS	HO6C-TI	138.0	5.873	4.203	
49	HINO MOTORS	HO6C-T	113.0	6.669	4.823	
49	HINO MOTORS	EF750	243.0	7.474	4.815	
49	HINO MOTORS	WO4C	77.5	4.848	4.052	
49	HINO MOTORS	HO7C	132.0	11.504	7.424	
49	HINO MOTORS	WO6D	87.0	6.374	4.828	
49	HINO MOTORS	EM100	163.0	5.962	4.540	
49	HINO MOTORS	EF750T	243.8	10.277	11.278	0.214
49	HINO MOTORS	EK200	198.0	6.428	5.202	
49	HINO MOTORS	EK100	153.0	9.193	6.405	
49	HINO MOTORS	EF750	186.0	11.876	6.290	
49	HINO MOTORS	WO4D	62.0	6.966	5.484	
49	HINO MOTORS	EP100-TI	214.0	5.940	4.028	
49	HINO MOTORS	HO6C-T	129.8	5.805	9.249	0.216
49	HINO MOTORS	EF750T	214.0	11.690	5.607	
49	HINO MOTORS	WO6E	121.0	4.046	3.471	
49	ISUZU MOTORS LIMITED	6BG1T	127.5	5.696	3.960	
49	ISUZU MOTORS LIMITED	UM6BD1MTC3	157.3	5.343	4.029	
49	ISUZU MOTORS LIMITED	QD-40	30.6	8.855	15.635	0.271
49	ISUZU MOTORS LIMITED	4HF1	99.0		3.384	
49	ISUZU MOTORS LIMITED	6BG1TC	147.0	6.624	3.878	
49	ISUZU MOTORS LIMITED	6BD1	113.3	6.234	4.102	
49	ISUZU MOTORS LIMITED	6BD1T	123.0	5.948	8.599	
49	ISUZU MOTORS LIMITED	2KC1	11.3	12.368	8.437	
49	ISUZU MOTORS LIMITED	QD-60	41.8	7.087	11.758	0.268
49	ISUZU MOTORS LIMITED	6WA1TC	280.0		3.857	
49	ISUZU MOTORS LIMITED	6SA1T	145.4	5.973	4.264	
49	ISUZU MOTORS LIMITED	4BD1T	76.1	6.390	4.299	
49	ISUZU MOTORS LIMITED	6BD1T	115.6	5.253	4.283	
49	ISUZU MOTORS LIMITED	6RB1	168.5	7.179	5.696	
49	ISUZU MOTORS LIMITED	4JB1	43.3	8.152	5.803	

Four-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
49	ISUZU MOTORS LIMITED	6RB1T	196.9	7.571	5.435	
49	ISUZU MOTORS LIMITED	4BD1	61.9	7.106	5.203	
49	ISUZU MOTORS LIMITED	3KC1	17.5	9.399	5.878	
49	ISUZU MOTORS LIMITED	3KR1	22.0	9.595	6.001	
49	ISUZU MOTORS LIMITED	6BD1	96.9	6.740	4.694	
49	ISUZU MOTORS LIMITED	QD-90	58.2	7.938	11.932	0.225
49	ISUZU MOTORS LIMITED	C240	43.3	8.130	6.243	
49	ISUZU MOTORS LIMITED	C240	33.6	8.852	6.646	
49	ISUZU MOTORS LIMITED	3AB1	25.4	10.745	8.559	
49	ISUZU MOTORS LIMITED	6RA1T	210.0	9.293	4.929	0.213
49	ISUZU MOTORS LIMITED	6SA1T	164.1	4.917	3.444	0.226
49	ISUZU MOTORS LIMITED	6HE1-N	121.0		4.207	0.230
49	ISUZU MOTORS LIMITED	6HE1-S	143.0		3.559	0.230
49	ISUZU MOTORS LIMITED	6SA1	117.1	7.136	5.040	
49	ISUZU MOTORS LIMITED	6BB1	83.5	7.824	5.448	
49	ISUZU MOTORS LTD.	1.817 L 4FB1	38.0	7.127	4.579	
49	JOHN DEERE	6068D	97.0		6.062	
49	JOHN DEERE	6059D	89.0		5.820	
49	JOHN DEERE	3179D	43.0	6.698	7.512	
49	JOHN DEERE	6076H	183.0		4.339	
49	JOHN DEERE	4039D	60.0		7.033	
49	JOHN DEERE	6059T	123.0		4.268	
49	JOHN DEERE	4045T	86.0		5.663	
49	JOHN DEERE	4239A	87.0		5.253	
49	JOHN DEERE	4039T	82.0		5.329	
49	JOHN DEERE	6619A	225.0		4.929	
49	JOHN DEERE	6076A	168.0		4.946	
49	JOHN DEERE	3179T	59.0		5.593	
49	JOHN DEERE	4045D	63.0		7.524	
49	JOHN DEERE	6359A	131.0	4.350	4.733	
49	JOHN DEERE	6068T	130.0		4.631	
49	JOHN DEERE	6466A	168.5	4.928	10.681	0.216
49	KOMATSU	SA6D110-1	164.1	5.758	8.065	0.206
49	KOMATSU	6D125-1	152.9	6.446	12.187	0.211
49	KOMATSU	SA6D140	368.0	3.885	3.342	
49	KOMATSU	SA12V170	1102.9	4.248	8.755	
49	KOMATSU	S6D140	294.0	4.863	4.082	
49	KOMATSU	SA8V140	459.3	3.656	7.384	
49	KOMATSU	SA6D125-1	275.9	3.884	7.191	0.196
49	KUBOTA	D1105-B	18.6		6.222	
49	KUBOTA	D1005-B	19.4		5.983	
49	KUBOTA	V1205-B	23.5		5.662	
49	KUBOTA	D3200B	49.2	6.683	14.122	0.228
49	KUBOTA	V1505-B	24.6		5.405	
49	KUBOTA	WG750-B	17.2		3.597	
49	KUBOTA	D905-B	17.5		6.620	
49	KUBOTA	V4300B	65.6	6.124	12.816	0.225
49	KUBOTA	V1902B	31.3	7.422	13.666	0.262
49	KUBOTA	V1305-B	25.7		5.170	
49	KUBOTA	WG600-B	14.2		4.355	
49	M.A.N.	D2566MK	237.9	7.595	7.878	
49	M.A.N.	DO226MKF	146.2	5.041	6.486	

Four-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
49	MACK	E6-350-4VHCMCAC	261.0		7.989	0.206
49	MITSUBISHI MOTORS CORP.	4D31T	89.5	4.555	7.264	
49	MITSUBISHI MOTORS CORP.	6D14	115.6	5.065	9.820	
49	MITSUBISHI MOTORS CORP.	6D16	136.1	5.323	3.584	
49	MITSUBISHI MOTORS CORP.	6D22T3	246.1	7.593	8.550	
49	MITSUBISHI MOTORS CORP.	6D22	167.8	12.176	11.759	
49	MITSUBISHI MOTORS CORP.	6D14T	145.4	4.928	8.259	
49	MWM	D 226-4	62.6	5.462	12.500	
49	MWM	D 226.6T	114.1	5.063	9.221	
49	NAVISTAR	T 444E: HEUI	160.3	3.121		
49	NAVISTAR	DT-239	67.1	6.521	13.783	0.253
49	NAVISTAR	6.9 L	126.8	5.180	6.705	
49	NISSAN DIESEL	SD1604	25.0		7.197	
49	NISSAN DIESEL	PE6	151.4	3.691	12.089	
49	NISSAN DIESEL	SD33	65.6	7.269	10.073	
49	NISSAN DIESEL	PE6-TA	216.3		10.173	
49	NISSAN DIESEL	SD226J	34.6		8.678	
49	NISSAN DIESEL	TD2704	51.5		4.855	
49	NISSAN DIESEL	BD3004	55.2		4.714	
49	NISSAN DIESEL	TD2304	44.1		5.665	
49	NISSAN DIESEL	ND6	97.7	9.528	12.868	
49	NISSAN DIESEL	FD3504	62.5		4.878	
49	NISSAN DIESEL	NE6	130.5		4.904	
49	NISSAN DIESEL	SD3304	53.0		5.665	
49	NISSAN DIESEL	SD336J	53.7		7.171	
49	NISSAN DIESEL	SD22	44.7	5.612	10.594	
49	NISSAN DIESEL	FD3304	58.8		5.150	
49	NISSAN DIESEL	FD35TA16	95.6		6.223	
49	NISSAN DIESEL	FD614	84.6		5.403	
49	NISSAN DIESEL	SD2204	34.6		6.219	
49	NISSAN DIESEL	FD35T04	77.2		4.208	
49	NISSAN DIESEL	FD33T04	73.6		4.351	
49	NISSAN DIESEL	SD33T6	61.8		8.498	
49	NISSAN DIESEL	SD2504	39.7		5.539	
49	NISSAN DIESEL	ND6T	120.8	15.433	11.225	
49	NISSAN DIESEL	FD606	84.6		8.631	
49	NISSAN DIESEL	NE6T	156.6		4.246	
49	NISSAN DIESEL	FE6	126.8		4.023	
49	NISSAN DIESEL	SD33T	78.3	5.281	8.289	
49	PERKINS	4.2032	44.0	7.980	11.728	0.255
49	PERKINS	4.108	32.8	6.154	15.544	0.257
49	PERKINS	T6.60 CC	134.2	4.498	8.568	
49	PERKINS	TV8.640	187.2	4.564	10.258	
49	PERKINS	T4.40 CC	89.5	5.454	8.694	
49	PEUGEOT SA	2.3 L XD2S	60.7		3.542	
49	PEUGEOT SA	2.3 L XD2	51.5		3.883	
49	POYAUD	060212	131.2	3.553	8.648	
49	POYAUD	062030S	175.2	6.072	10.985	
49	RENAULT	062045 MIDR	220.0	4.965	8.819	
49	RENAULT	063540	231.2	6.301	10.962	
49	RENAULT	720	65.6	5.047	10.408	
49	SCANIA	DSC11.03	254.3		8.585	0.204

Four-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
49	SCANIA	DSC9	211.8	5.198	8.330	
49	SCANIA	DS14.01	324.4	5.456	8.086	0.209
49	SSANGYONG HEAVY IND.	WARTSILA 12V32	4920.0	15.618	15.152	
49	SSANGYONG HEAVY IND.	MAN B&W 9L28/32H	1980.0	21.668	12.741	
49	SSANGYONG HEAVY IND.	WARTSILA 12V22	2100.0	9.405	13.074	
49	SSANGYONG HEAVY IND.	WARTSILA 8V22	1400.0	8.691	12.662	
49	SSANGYONG HEAVY IND.	WARTSILA 18V32	7380.0	11.890	6.898	
49	SSANGYONG HEAVY IND.	WARTSILA 16V32	6560.0	14.345	12.611	
49	SSANGYONG HEAVY IND.	WARTSILA 6R22/26	1125.0	12.819	7.515	
49	SSANGYONG HEAVY IND.	WARTSILA 16V32	6560.0	11.315	7.206	
49	SSANGYONG HEAVY IND.	WARTSILA 6R22/26	1065.0	12.475	14.938	
49	SSANGYONG HEAVY IND.	MAN B&W 6L28/32H	1320.0	19.494	14.291	
49	SSANGYONG HEAVY IND.	WARTSILA 16V22	2800.0	10.434	6.591	
49	SSANGYONG HEAVY IND.	MAN B&W 8L28/32H	1760.0	21.968	12.887	
49	SSANGYONG HEAVY IND.	WARTSILA 4R22/26	710.0	15.274	19.462	
49	SSANGYONG HEAVY IND.	WARTSILA 4R32	1640.0	16.340	18.847	
49	SSANGYONG HEAVY IND.	WARTSILA 9R32	3690.0	16.479	17.246	
49	SSANGYONG HEAVY IND.	WARTSILA 8R22/26	1500.0	11.384	6.848	
49	SSANGYONG HEAVY IND.	MAN B&W 5L23/30	800.0	20.591	13.807	
49	SSANGYONG HEAVY IND.	MAN B&W T23LH-4E	550.0	14.274	16.033	
49	SSANGYONG HEAVY IND.	WARTSILA 8R22	1400.0	12.197	7.338	
49	SSANGYONG HEAVY IND.	WARTSILA 8R32	3280.0	17.068	17.461	
49	SSANGYONG HEAVY IND.	MAN B&W 7L28/32H	1540.0	22.099	13.312	
49	SSANGYONG HEAVY IND.	WARTSILA 8R22	1400.0	14.970	13.506	
49	SSANGYONG HEAVY IND.	WARTSILA 8V22	1400.0	12.838	6.558	
49	SSANGYONG HEAVY IND.	WARTSILA 4R22	700.0	16.811	9.870	
49	SSANGYONG HEAVY IND.	WARTSILA 16V22	2800.0	8.948	12.240	
49	SSANGYONG HEAVY IND.	WARTSILA 8R32	3280.0	15.361	9.839	
49	SSANGYONG HEAVY IND.	WARTSILA 9R32	3690.0	16.386	9.855	
49	SSANGYONG HEAVY IND.	MAN B&W 8L23/30	1280.0	20.516	11.222	
49	SSANGYONG HEAVY IND.	WARTSILA 6R32	2460.0	17.454	9.608	
49	SSANGYONG HEAVY IND.	WARTSILA 4R22	700.0	17.403	19.740	
49	SSANGYONG HEAVY IND.	WARTSILA 12V22	2100.0	11.864	6.667	
49	SSANGYONG HEAVY IND.	MAN B&W 7L23/30	1120.0	17.929	11.607	
49	SSANGYONG HEAVY IND.	MAN B&W 6L23/30	960.0	19.038	12.216	
49	SSANGYONG HEAVY IND.	WARTSILA 6R22	1050.0	13.735	8.052	
49	SSANGYONG HEAVY IND.	WARTSILA 4R32	1640.0	20.008	10.255	
49	SSANGYONG HEAVY IND.	WARTSILA 4R22/26	750.0	15.690	9.212	
49	SSANGYONG HEAVY IND.	WARTSILA 18V32	7380.0	14.223	12.318	
49	SSANGYONG HEAVY IND.	SULZER 9S20	1440.0	17.022	8.144	
49	SSANGYONG HEAVY IND.	WARTSILA 6R32	2460.0	14.485	16.630	
49	SSANGYONG HEAVY IND.	SULZER 8S20	1280.0	17.763	8.381	
49	SSANGYONG HEAVY IND.	MAN B&W 18V28/32H	3960.0	15.790	10.813	
49	SSANGYONG HEAVY IND.	SULZER 6S20	960.0	18.384	8.902	
49	SSANGYONG HEAVY IND.	WARTSILA 6R22	1050.0	14.304	15.152	
49	SSANGYONG HEAVY IND.	WARTSILA 8R22/26	1420.0	13.000	13.316	
49	SSANGYONG HEAVY IND.	MAN B&W 12V28/32H	3585.0	9.679	8.381	
49	SSANGYONG HEAVY IND.	MAN B&W 5L28/32H	1100.0	21.004	15.000	
49	SSANGYONG HEAVY IND.	MAN B&W 16V28/32H	4785.0	11.081	8.673	
49	SSANGYONG HEAVY IND.	WARTSILA 12V32	4920.0	13.802	7.945	
49	STEYR	610	99.2	6.358	11.999	0.231
49	STEYR	615.68	230.4	3.575	7.725	

Four-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
49	STEYR	815.67	249.1	5.250	9.207	
49	VOLKSWAGEN AG	074.Y	53.0	4.565	3.302	
49	VOLKSWAGEN AG	068.D	37.0	5.548	3.486	
49	VOLKSWAGEN AG	028.B	45.0	4.586	2.889	
49	VOLVO	TD 121F	283.4	5.376	8.470	
49	VOLVO	TD 61 F	152.1	6.322	8.414	
49	VOLVO	TD 101F	220.0	4.881	9.910	
49	WARTSILA	9R20	1170.0	10.999	10.085	
49	WARTSILA	4R20	520.0	14.229	12.308	
49	YANMAR	3TNA72E	15.7	6.610	5.428	
49	YANMAR	4TN82E	35.0	5.555	5.193	
49	YANMAR	3TNC78C	22.4	6.373	5.141	
49	YANMAR	3TN66E	12.3	7.525	5.039	
49	YANMAR	4TN82TE	41.0	5.407	4.560	
49	YANMAR	4TN84E	36.5	5.329	4.926	
49	YANMAR	3TN84E	27.2	6.141	5.327	
49	YANMAR	3TN100E	44.2	5.659	4.749	
49	YANMAR	2TN66E	8.2	7.758	6.098	
49	YANMAR	4TN84TE	42.5	5.217	4.352	
49	YANMAR	3TN82E	26.1	6.404	5.556	
49	YANMAR	3TN75E	18.3	7.803	6.568	
49	YANMAR	3TNC78E	22.4	6.373	5.141	
49	YANMAR	3TN82TE	29.1	6.457	5.158	
49	YANMAR	4TN100E	57.1	5.171	4.377	
49	YANMAR	4TN100TE	74.2	4.294	3.437	
51	Audi	2.5-L	88.0			0.198
52	Ford	1.8-L	55.0			0.255
53	Mitsubishi	4M40	69.0			0.249
53	Mitsubishi	4M40-TI	92.0			0.249

Two-Stroke Spark-Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
54	Peugeot	IAPAC SCRE	28.00			0.288
16	Confidential	Confidential	77.00		0.909	
16	Confidential	Confidential	82.00		0.915	
55	URM	500	29.75		1.143	
55	QUB	500	48.12			0.358

Two-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
17	SwRI	UAV	22.37	2.079	0.710	0.247
49	DETROIT DIESEL	2-71	47.72	12.164	20.116	0.285
49	DETROIT DIESEL	3-53N	73.08	7.078	13.205	0.279
49	DETROIT DIESEL	4-53N	101.60	6.252	10.925	0.269
49	DETROIT DIESEL	6-71N	186.60	7.141	11.522	0.254
49	DETROIT DIESEL	12V-92N	397.20	6.630	9.844	0.252
49	DETROIT DIESEL	12V-149N	592.80	9.252	14.322	0.249
49	DETROIT DIESEL	16V-149	797.89	7.320	13.147	0.249
49	DETROIT DIESEL	8V-71N	248.80	6.157	9.285	0.246
49	DETROIT DIESEL	4-71N	124.40	6.739	14.309	0.246
49	DETROIT DIESEL	6V-92N	217.20	5.670	9.024	0.241
49	DETROIT DIESEL	8V-92N	289.60	5.402	8.097	0.240
49	DETROIT DIESEL	4-53T	137.95	5.232	9.134	0.237
49	DETROIT DIESEL	3-53T	104.40	6.022	9.579	0.234
49	DETROIT DIESEL	8V-149TI	596.80	7.648	10.054	0.231
49	DETROIT DIESEL	12V-149TIB	1006.68	6.383	9.000	0.224
49	DETROIT DIESEL	4-71T	157.34	6.357	11.631	0.221
49	DETROIT DIESEL	16V-92TA	715.86	5.015	6.761	0.221
49	DETROIT DIESEL	6V-53T	238.80	4.060	7.098	0.220
49	DETROIT DIESEL	16V-149TIB	1342.24	5.632	8.352	0.218
49	DETROIT DIESEL	8V-92TA	357.93	3.741	6.761	0.217
49	DETROIT DIESEL	12V-92TA	805.92	3.509	5.311	0.215
49	DETROIT DIESEL	8V-71TA	298.28	5.822	8.365	0.214
49	DETROIT DIESEL	6V-92TA	354.60	3.136	5.697	0.212
49	DETROIT DIESEL	6-71TA	324.60	4.690	6.762	0.209
49	DIESEL UNITED, LTD.	7RTA84T	27184.07	37.980		
49	DIESEL UNITED, LTD.	7RTA52	9938.60	23.439		
49	DIESEL UNITED, LTD.	9RTA84T	34950.94	38.181		
49	DIESEL UNITED, LTD.	7RTA76	18946.47	34.462		
49	DIESEL UNITED, LTD.	10RTA84C	38245.98	30.335		
49	DIESEL UNITED, LTD.	5RTA52	7097.57	25.686		
49	DIESEL UNITED, LTD.	6RTA52	8517.09	24.375		
49	DIESEL UNITED, LTD.	12RTA84M	44747.80	35.529		
49	DIESEL UNITED, LTD.	6RTA62	12179.87	27.839		
49	DIESEL UNITED, LTD.	4RTA52	5678.06	27.652		
49	DIESEL UNITED, LTD.	6RTA76	16239.83	35.626		
49	DIESEL UNITED, LTD.	7RTA62	14209.85	26.791		
49	DIESEL UNITED, LTD.	8RTA62	16239.83	26.006		
49	DIESEL UNITED, LTD.	8RTA84M	29831.86	38.705		
49	DIESEL UNITED, LTD.	7RTA84M	26102.88	37.461		
49	DIESEL UNITED, LTD.	5RTA76	13533.19	37.256		
49	DIESEL UNITED, LTD.	8RTA76	21653.11	33.589		
49	DIESEL UNITED, LTD.	10RTA76	27066.39	34.139		
49	DIESEL UNITED, LTD.	5RTA84C	19122.99	33.599		
49	DIESEL UNITED, LTD.	6RTA72	16493.23	31.679		
49	DIESEL UNITED, LTD.	4RTA62	8119.92	31.505		
49	DIESEL UNITED, LTD.	8RTA84C	30596.78	31.903		
49	DIESEL UNITED, LTD.	8RTA52	11429.66	22.590		
49	DIESEL UNITED, LTD.	9RTA84C	34421.38	31.032		
49	DIESEL UNITED, LTD.	5RTA72	13717.07	33.375		
49	DIESEL UNITED, LTD.	4RTA84C	15298.39	35.982		
49	DIESEL UNITED, LTD.	12RTA84C	45895.18	29.290		

Two-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
49	DIESEL UNITED, LTD.	8RTA72	21947.31	29.701		
49	DIESEL UNITED, LTD.	7RTA72	19203.90	30.576		
49	DIESEL UNITED, LTD.	7RTA84C	26772.19	30.874		
49	DIESEL UNITED, LTD.	9RTA84M	33538.78	37.671		
49	DIESEL UNITED, LTD.	6RTA84C	22947.59	32.009		
49	DIESEL UNITED, LTD.	12RTA76	32479.66	33.028		
49	DIESEL UNITED, LTD.	5RTA62	10149.89	29.305		
49	DIESEL UNITED, LTD.	10RTA84M	37289.83	38.800		
49	SSANGYONG HEAVY INI	MAN B&W L35MC	3920.00		17.625	
49	FAIRBANKS MORSE EN(	PC2.5	6530.00	14.191	11.593	
49	FAIRBANKS MORSE EN(	38ETDD8-1/8	3165.00	15.272	12.351	
49	FAIRBANKS MORSE EN(	PC2.5	5600.00	14.979	11.786	
49	SSANGYONG HEAVY INI	MAN B&W L35MC	3360.00		17.857	
49	FAIRBANKS MORSE EN(	PC2.5	8400.00	17.085	10.714	
49	FAIRBANKS MORSE EN(	38ETD8-1/8	1580.00	23.142	17.837	
49	FAIRBANKS MORSE EN(	PC2.5	7465.00	15.143	11.119	
49	SSANGYONG HEAVY INI	MAN B&W L35MC	4480.00		17.045	
49	FAIRBANKS MORSE EN(	38ETD8-1/8	2370.00	19.975	14.384	
49	FAIRBANKS MORSE EN(	38ETDS8-1/8	2585.00	18.726	15.122	
49	SSANGYONG HEAVY INI	MAN B&W S26MC	1825.00		14.944	
49	SSANGYONG HEAVY INI	MAN B&W S26MC	2190.00		14.529	
49	SSANGYONG HEAVY INI	MAN B&W S26MC	2920.00		14.010	
49	FAIRBANKS MORSE EN(	38ETDS8-1/8	2150.00	20.816	16.490	
49	SSANGYONG HEAVY INI	MAN B&W S26MC	1460.00		15.567	
49	SSANGYONG HEAVY INI	MAN B&W S26MC	2555.00		14.232	
49	FAIRBANKS MORSE EN(	38ETDS8-1/8	1720.00	19.984	18.499	
49	DIESEL UNITED, LTD.	4RTA76	10826.55	39.700		
49	DIESEL UNITED, LTD.	4RTA84M	14915.93	43.674		
49	DIESEL UNITED, LTD.	4RTA72	10973.65	35.824		
49	DIESEL UNITED, LTD.	8RTA84T	31067.50	39.308		
49	DIESEL UNITED, LTD.	6RTA84M	22373.90	38.842		
49	DIESEL UNITED, LTD.	9RTA76	24359.75	35.089		
49	DIESEL UNITED, LTD.	5RTA84T	19417.19	41.508		
49	DIESEL UNITED, LTD.	5RTA84M	18644.92	40.775		
49	DIESEL UNITED, LTD.	6RTA84T	23300.63	39.450		
49	SSANGYONG HEAVY INI	MAN B&W L35MCE	3600.00		21.212	
49	SSANGYONG HEAVY INI	MAN B&W L35MCE	3150.00		21.934	
49	SSANGYONG HEAVY INI	MAN B&W L35MC	2800.00		18.831	
49	SSANGYONG HEAVY INI	MAN B&W L35MC	2240.00		20.698	
49	SSANGYONG HEAVY INI	MAN B&W L35MCE	2250.00		23.434	
49	SSANGYONG HEAVY INI	MAN B&W L35MCE	1800.00		25.758	
49	SSANGYONG HEAVY INI	MAN B&W L35MCE	2700.00		22.222	
49	FAIRBANKS MORSE EN(	38ETDS8-1/8	1240.00	28.296	22.727	

Rotary Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
1	Predicted best attainable		30.00	1.000	1.300	0.300
24	Alvis	AR731	28.30	0.360	0.350	0.316
2	Norton		30.00	1.400	0.700	0.300
2	Rotac		25.00	1.200	1.100	0.360
56	Yanmar	R220	16.55			0.382
56	Yanmar	R450	36.78			0.382
57	Norton, AAI Modifications	NR631	28.50		1.195	0.320
58	JTDI Dem/Val	2116 R	560.00			0.230
58	JTDI	580 Series	846.37			0.224
58	JTDI	580 Series	149.14			0.268
58	JTDI	580 Series	300.00		0.760	0.237

Two-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
17	SwRI	UAV	22.37	2.079	0.710	0.247
49	DETROIT DIESEL	2-71	47.72	12.164	20.116	0.285
49	DETROIT DIESEL	3-53N	73.08	7.078	13.205	0.279
49	DETROIT DIESEL	4-53N	101.60	6.252	10.925	0.269
49	DETROIT DIESEL	6-71N	186.60	7.141	11.522	0.254
49	DETROIT DIESEL	12V-92N	397.20	6.630	9.844	0.252
49	DETROIT DIESEL	12V-149N	592.80	9.252	14.322	0.249
49	DETROIT DIESEL	16V-149	797.89	7.320	13.147	0.249
49	DETROIT DIESEL	8V-71N	248.80	6.157	9.285	0.246
49	DETROIT DIESEL	4-71N	124.40	6.739	14.309	0.246
49	DETROIT DIESEL	6V-92N	217.20	5.670	9.024	0.241
49	DETROIT DIESEL	8V-92N	289.60	5.402	8.097	0.240
49	DETROIT DIESEL	4-53T	137.95	5.232	9.134	0.237
49	DETROIT DIESEL	3-53T	104.40	6.022	9.579	0.234
49	DETROIT DIESEL	8V-149TI	596.80	7.648	10.054	0.231
49	DETROIT DIESEL	12V-149TIB	1006.68	6.383	9.000	0.224
49	DETROIT DIESEL	4-71T	157.34	6.357	11.631	0.221
49	DETROIT DIESEL	16V-92TA	715.86	5.015	6.761	0.221
49	DETROIT DIESEL	6V-53T	238.80	4.060	7.098	0.220
49	DETROIT DIESEL	16V-149TIB	1342.24	5.632	8.352	0.218
49	DETROIT DIESEL	8V-92TA	357.93	3.741	6.761	0.217
49	DETROIT DIESEL	12V-92TA	805.92	3.509	5.311	0.215
49	DETROIT DIESEL	8V-71TA	298.28	5.822	8.365	0.214
49	DETROIT DIESEL	6V-92TA	354.60	3.136	5.697	0.212
49	DETROIT DIESEL	6-71TA	324.60	4.690	6.762	0.209
49	DIESEL UNITED, LTD.	7RTA84T	27184.07	37.980		
49	DIESEL UNITED, LTD.	7RTA52	9936.60	23.439		
49	DIESEL UNITED, LTD.	9RTA84T	34950.94	38.181		
49	DIESEL UNITED, LTD.	7RTA76	18946.47	34.462		
49	DIESEL UNITED, LTD.	10RTA84C	38245.98	30.335		
49	DIESEL UNITED, LTD.	5RTA52	7097.57	25.686		
49	DIESEL UNITED, LTD.	6RTA52	8517.09	24.375		
49	DIESEL UNITED, LTD.	12RTA84M	44747.80	35.529		
49	DIESEL UNITED, LTD.	6RTA62	12179.87	27.839		
49	DIESEL UNITED, LTD.	4RTA52	5678.06	27.652		
49	DIESEL UNITED, LTD.	6RTA76	16239.83	35.626		
49	DIESEL UNITED, LTD.	7RTA62	14209.85	26.791		
49	DIESEL UNITED, LTD.	8RTA62	16239.83	26.006		
49	DIESEL UNITED, LTD.	8RTA84M	29831.86	38.705		
49	DIESEL UNITED, LTD.	7RTA84M	26102.88	37.461		
49	DIESEL UNITED, LTD.	5RTA76	13533.19	37.256		
49	DIESEL UNITED, LTD.	8RTA76	21653.11	33.589		
49	DIESEL UNITED, LTD.	10RTA76	27066.39	34.139		
49	DIESEL UNITED, LTD.	5RTA84C	19122.99	33.599		
49	DIESEL UNITED, LTD.	6RTA72	16493.23	31.679		
49	DIESEL UNITED, LTD.	4RTA62	8119.92	31.505		
49	DIESEL UNITED, LTD.	8RTA84C	30596.78	31.903		
49	DIESEL UNITED, LTD.	8RTA52	11429.66	22.590		
49	DIESEL UNITED, LTD.	9RTA84C	34421.38	31.032		
49	DIESEL UNITED, LTD.	5RTA72	13717.07	33.375		
49	DIESEL UNITED, LTD.	4RTA84C	15298.39	35.982		
49	DIESEL UNITED, LTD.	12RTA84C	45895.18	29.290		

Two-Stroke Compression Ignition Engines

Reference	Mfg	Model	Rated kW	Spec. Vol. l/kW	Spec. Wt. kg/kW	Min BSFC kg/kWh
49	DIESEL UNITED, LTD.	8RTA72	21947.31	29.701		
49	DIESEL UNITED, LTD.	7RTA72	19203.90	30.576		
49	DIESEL UNITED, LTD.	7RTA84C	26772.19	30.874		
49	DIESEL UNITED, LTD.	9RTA84M	33538.78	37.671		
49	DIESEL UNITED, LTD.	6RTA84C	22947.59	32.009		
49	DIESEL UNITED, LTD.	12RTA76	32479.66	33.028		
49	DIESEL UNITED, LTD.	5RTA62	10149.89	29.305		
49	DIESEL UNITED, LTD.	10RTA84M	37289.83	36.800		
49	SSANGYONG HEAVY IND.	MAN B&W L35MC	3920.00		17.625	
49	FAIRBANKS MORSE ENGINES	PC2.5	6530.00	14.191	11.593	
49	FAIRBANKS MORSE ENGINES	38ETDD8-1/8	3165.00	15.272	12.351	
49	FAIRBANKS MORSE ENGINES	PC2.5	5600.00	14.979	11.786	
49	SSANGYONG HEAVY IND.	MAN B&W L35MC	3360.00		17.857	
49	FAIRBANKS MORSE ENGINES	PC2.5	8400.00	17.085	10.714	
49	FAIRBANKS MORSE ENGINES	38ETD8-1/8	1580.00	23.142	17.837	
49	FAIRBANKS MORSE ENGINES	PC2.5	7465.00	15.143	11.119	
49	SSANGYONG HEAVY IND.	MAN B&W L35MC	4480.00		17.045	
49	FAIRBANKS MORSE ENGINES	38ETD8-1/8	2370.00	19.975	14.384	
49	FAIRBANKS MORSE ENGINES	38ETDS8-1/8	2585.00	18.726	15.122	
49	SSANGYONG HEAVY IND.	MAN B&W S26MC	1825.00		14.944	
49	SSANGYONG HEAVY IND.	MAN B&W S26MC	2190.00		14.529	
49	SSANGYONG HEAVY IND.	MAN B&W S26MC	2920.00		14.010	
49	FAIRBANKS MORSE ENGINES	38ETDS8-1/8	2150.00	20.816	16.490	
49	SSANGYONG HEAVY IND.	MAN B&W S26MC	1460.00		15.567	
49	SSANGYONG HEAVY IND.	MAN B&W S26MC	2555.00		14.232	
49	FAIRBANKS MORSE ENGINES	38ETDS8-1/8	1720.00	19.984	18.499	
49	DIESEL UNITED, LTD.	4RTA76	10826.55	39.700		
49	DIESEL UNITED, LTD.	4RTA84M	14915.93	43.674		
49	DIESEL UNITED, LTD.	4RTA72	10973.65	35.824		
49	DIESEL UNITED, LTD.	8RTA84T	31067.50	39.308		
49	DIESEL UNITED, LTD.	6RTA84M	22373.90	38.842		
49	DIESEL UNITED, LTD.	9RTA76	24359.75	35.089		
49	DIESEL UNITED, LTD.	5RTA84T	19417.19	41.508		
49	DIESEL UNITED, LTD.	5RTA84M	18644.92	40.775		
49	DIESEL UNITED, LTD.	6RTA84T	23300.63	39.450		
49	SSANGYONG HEAVY IND.	MAN B&W L35MCE	3600.00		21.212	
49	SSANGYONG HEAVY IND.	MAN B&W L35MCE	3150.00		21.934	
49	SSANGYONG HEAVY IND.	MAN B&W L35MC	2800.00		18.831	
49	SSANGYONG HEAVY IND.	MAN B&W L35MC	2240.00		20.698	
49	SSANGYONG HEAVY IND.	MAN B&W L35MCE	2250.00		23.434	
49	SSANGYONG HEAVY IND.	MAN B&W L35MCE	1800.00		25.758	
49	SSANGYONG HEAVY IND.	MAN B&W L35MCE	2700.00		22.222	
49	FAIRBANKS MORSE ENGINES	38ETDS8-1/8	1240.00	28.296	22.727	

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